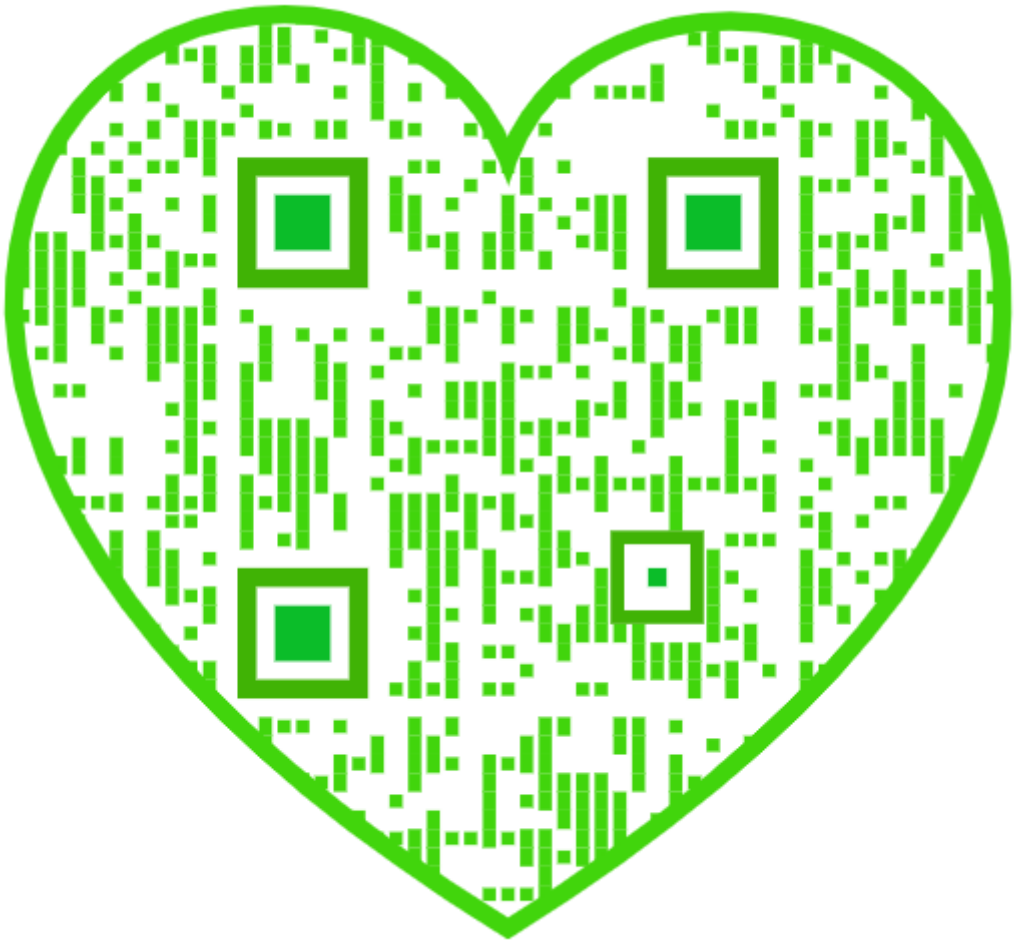
REFRIGERATION
IN THE
CHEMICAL INDUSTRY

G. W. DANIELS



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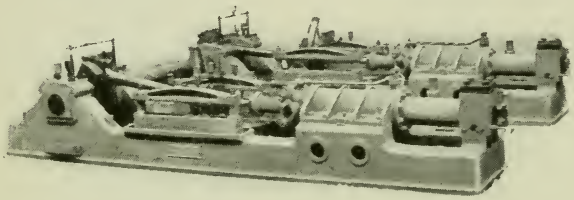

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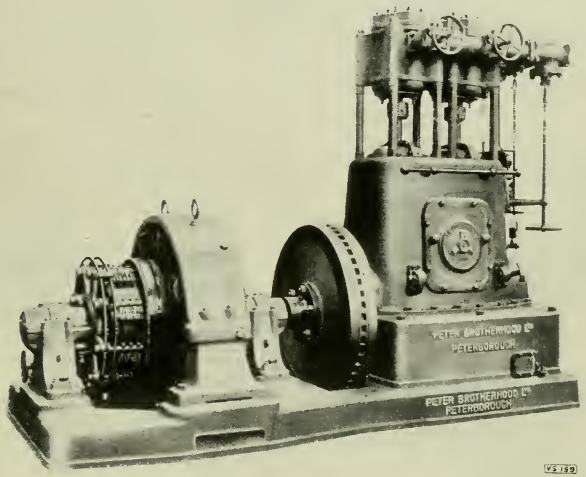

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BY
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WITH THIRTY-FIVE ILLUSTRATIONS,
FOURTEEN TABLES, AND APPENDIX

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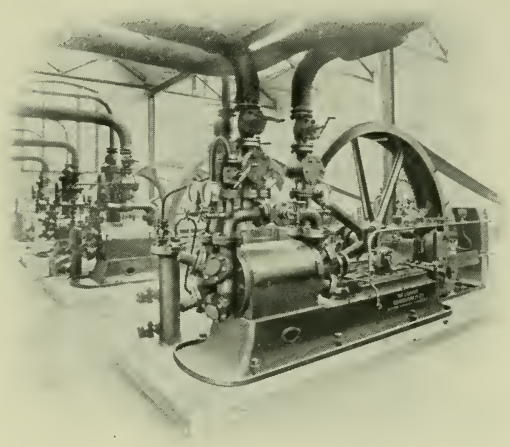

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PREFACE.

In writing this volume I have been influenced by my experience of the chemical industry, in which an ever-increasing use is made of refrigeration. I have found that many chemists and chemical engineers have no appreciation at all of the subject, either qualitatively or quantitatively. While therefore the chemical works staffs are not in general called upon to design the actual refrigerating machinery, they have frequently to design the apparatus in which the cold is *applied* to the particular process under consideration. In any case, they should know the principles underlying the construction and working of the machinery, and have some idea of the amount of refrigeration obtainable under various conditions from machinery of given size, and the power required to operate it.

The scope of this volume is therefore arranged to enable the chemical engineer and chemist to form some correct quantitative idea of the capabilities of refrigerating plant, of the amount of refrigeration necessary for a given purpose, and to apply the refrigeration when produced.

It assumes some elementary knowledge of engineering construction and materials, and of physics, mechanics, etc.

To obtain a good working knowledge of refrigeration, it is not necessary to elaborate the thermodynamical part of the subject. Those engineers who wish to specialize in refrigeration can readily obtain the knowledge elsewhere.

The production of liquid air and similar extremely low temperature work is not dealt with because there are sufficient treatises already available on such matters.

I am indebted to the following firms for assistance in providing illustrations.

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Messrs. J. and E. Hall, Ltd., Dartford	7, 14a, 19
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I have also to thank Messrs. Blackie and Co., Ltd., for permission to publish Figure 8, this being generally similar to an illustration in the general article on Refrigeration in their work, "Modern Mechanical Engineering."

The Author will be obliged if errors or omissions are brought to his notice.

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Refrigeration in Chemical Industry

Chapter I.

SOME FUNDAMENTAL POINTS.

Methods of Producing Low Temperatures.

Temperatures lower than atmospheric may be produced in five distinct ways, namely: —

(1) by the use of naturally formed ice;
(2) by freezing mixtures;
(3) by expansion of a gas through a throttle valve;
(4) by expansion of a gas doing external work;
(5) by evaporation of a liquid.

The first method is obviously restricted to regions in which there are sufficiently severe climatic conditions to produce the necessary quantity of ice. Considerable business is done in such ice in certain parts of America, Norway, and similar places, but such methods are of little or no interest to the chemical industry, nor do they need any explanation.

The second method is a laboratory method, and also needs no comment. Some freezing mixtures and the temperatures attainable with them are detailed in the Appendix.

The third method is only utilized in such processes as Linde's method of air liquefaction and has been fully dealt with elsewhere.

The last two methods are the most used for producing refrigeration on a commercial scale, and of these, the fifth process is the most important.

Cold Produced by Evaporating a Liquid.

Practically all refrigeration is now produced by the evaporation of a liquid, heat being removed from the immediate surroundings of the liquid, and absorbed by it to an extent depending on the latent heat of vaporization of the particular liquid used. In fact, the process of the artificial

production of cold is governed by three very simple physical laws: —

(1) Any liquid, in evaporating, abstracts heat from its surroundings.

(2) The temperature at which any liquid evaporates depends on the pressure to which it is subjected.

(3) Any vapour may be condensed to a liquid by subjecting it to a suitable combination of pressure and temperature.

The first law makes refrigeration possible, as we have already seen. It permits us to abstract heat, and therefore reduce the temperature of a body.

The second law enables us to control the low temperature or degree of refrigeration produced by the evaporating liquid. To do this we have to devise some means to vary at will the pressure under which the liquid evaporates.

The third law makes refrigeration a commercial possibility, in so much that the vapour formed when the liquid evaporates can be compressed and cooled in a condenser. It will then re-liquefy, and can be used over again to produce more refrigeration. If this were not done we should have continually to make or purchase fresh supplies of liquid, and such a course would be far too expensive for commercial operation.

The liquid which we evaporate to produce the cold is called the *refrigerant*.

While the foregoing simple facts explain the artificial production of cold by the evaporation of a liquid, it is desirable for us to consider briefly the thermodynamic principles which also underly the operation.

The Refrigerating Machine a Heat Pump.

To refrigerate any material or enclosed space, we have first to remove from it sufficient heat to reduce its temperature to the desired amount below its usual temperature. This done, heat will at once begin to flow into the cooled material or space from the immediate surroundings which are at a higher temperature.

We must, therefore, continue to remove heat if we desire to maintain the material, or space, continually refrigerated. The operation is comparable with that of pumping water, say, from a mine shaft. In such a case, the pump takes in water at some low level at or near the bottom of the shaft, and discharges it at some higher level, where it can run away and not trouble us any more. Similarly with the refrigerating machine. This takes in heat at some low level of temperature and discharges it at some higher level of temperature where the heat can be dissipated or run to waste. We do not mind what becomes of the heat removed from the refrigerated body so long as it has no chance of flowing back into it. It is important to note that the heat removed from the cold body must be discharged and carried away because we cannot accumulate heat in the system of the refrigerating machine, nor by the first law of thermodynamics can we destroy the heat. In practice the heat absorbed from the body to be cooled is given up to the condensing water when the vapour of the liquid condenses after being compressed.

From our analogy of pumping water from a mine we thus get the idea of the refrigerating machine being a heat pump, and pumping heat from one level of temperature to another.

By the second law of thermodynamics, it is impossible for the heat taken up at the low level of temperature to be raised to the higher level of temperature without the expenditure of energy, *i.e.*, power has to be supplied to work the refrigerating machine and compress the vapour. The energy so expended is converted into heat during the compression, and is carried away by the condensing water, together with the heat removed from the cold body.

Thus, if h = heat removed from the cold body,

w = heat equivalent of the energy spent in compressing the vapour,

H = heat rejected to the condenser,

then--

$$H = h + w.$$

Coefficient of Performance of Ideal Refrigerating Machine.

Now we take, as a measure of the efficiency of a refrigerating machine, the ratio h/w, that is, the amount of heat removed from the cold body divided by the amount of energy spent in removing it. This ratio is mostly greater than unity, and for this reason it is called the coefficient of performance and not the efficiency. It will be observed that the *coefficient of performance is the amount of refrigeration obtained per unit of energy expended.*

From thermodynamic principles, considering the refrigerating machine as a heat pump, it can be proved that a theoretically ideal or perfect refrigerator would have a coefficient of performance given by the expression—

$$\frac{T_2}{T_1 - T_2}$$

where—

T_1 = high level of temperature at which the heat is rejected to the condenser.

T_2 = low level of temperature at which the liquid refrigerant evaporates and removes heat from the body to be cooled.

These temperatures are measured on the absolute scale, and in the case of the ideal machine would be constant during its operation. For example, all the heat taken from the cooled body would be removed at a constant temperature T_2.

Consideration of the expression $\frac{T_2}{T_1 - T_2}$ will show that, for the greatest coefficient of performance, $T_1 - T_2$ must be as small as possible. To accomplish this we must keep T_1 as low as possible by using a plentiful supply of the coolest condensing water available, and we must have T_2, the temperature of the evaporating refrigerant, no lower than is necessary for the purpose desired.

It sometimes happens in practice that we find it desirable or necessary to depart from this counsel of perfection,

and sacrifice this coefficient of performance to obtain other ends, such as, for example, to obtain the greatest output or quantity of refrigeration from a given sized plant.

No actual refrigerating machine can in practice have a coefficient of performance equal to that of the ideal machine and given by the formula $\frac{T_2}{T_1 - T_2}$. The reasons for this we shall see shortly.

The Vapour Compression Plant.

A vapour compression plant is illustrated in diagrammatic form in Figure I.

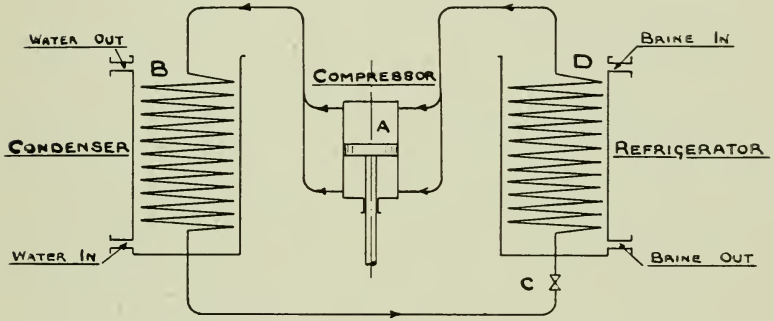

Figure 1

A is the compressor delivering the compressed vapour into the condensing coils *B*, which are cooled by water. The liquefied vapour then flows to the regulating valve *C* through which it is allowed to expand into the refrigerator coils *D*. Here it evaporates, abstracting heat from (and therefore cooling), say, brine surrounding the coil *D*. The vapour formed during the evaporation is removed by the compressor and re-liquefied for further use. The process is a continuous one.

It should be noted that the pressure under which the liquid evaporates in coil *D* can be varied by altering the speed of working of the compressor, and the amount of opening of the regulating valve *C*. Thus we have control of the temperature produced in the refrigerator.

Wet and Dry Compression.

We have already seen that work must be expended in pumping out the heat from the body to be cooled, and that this work is converted into an equivalent amount of heat. In consequence of this conversion of work into heat the vapour undergoes a rise of temperature during compression. We have therefore to see that this rise of temperature is not so great as to cause trouble in the working of the machine. An excessive temperature might, for example, cause difficulties with regard to the lubrication of the compressor, the prevention of leakage at the compressor rod stuffing box, deterioration of the refrigerating medium, and may also cause a reduction in the amount of refrigeration obtained from the machine. There are two methods adopted in practice to prevent this overheating.

The first is to provide the compressor with a water jacket.

The second is to so regulate the opening of the valve *C*, Figure 1, that rather more liquid is admitted to the refrigerator than can be evaporated. The balance is therefore carried in suspension by the vapour going to the compressor, and this liquid in suspension absorbs the heat generated during compression, by evaporating, and so prevents an excessive rise in temperature. The relative merits of these two methods will be further discussed in the chapter on machinery. We should note here, however, that if sufficient liquid is carried in suspension into the compressor to prevent, at any point in the compression, the vapour becoming superheated, we call such a method of operation one of *wet compression.* On the other hand, if no liquid at all is carried in suspension into the compressor, we say we are operating on a *dry compression system.* Both systems have their advocates, but the majority of machines in practice work on a mixture of the two, partly wet and partly dry.

A machine without a water jacket cannot be worked on the dry compression system, except under certain particular conditions of operation.

Loss at Regulating Valve.

It will be obvious that the liquid refrigerant leaving the condenser cannot be at a lower temperature than that of the cooling water supplied to the condenser, that is to say, at a temperature very much higher than that prevailing in the refrigerator. It follows that before such liquid refrigerant can produce any useful refrigeration, it must first itself be cooled down to the temperature in the refrigerator. In general, this cooling is accomplished by the expansion of the liquid as it passes through the regulating valve *C*, Figure 1. During this expansion a portion of the liquid evaporates, abstracting the heat necessary for this from the remaining liquid. This cooled remaining liquid then passes on to the refrigerator where it, in turn, evaporates and cools the brine surrounding the refrigerator coils. The amount of liquid refrigerant which has to be used up in this way to cool the remaining liquid to such a temperature as will enable it to produce useful refrigeration varies very much with the temperature conditions, that is, the tempera ture of the liquid approaching the regulating valve, and the temperature at which the liquid is evaporating in the refrigerator, and also upon the kind of refrigerant being used, e.g., ammonia, carbon dioxide, etc. In any case it constitutes a loss of some of the refrigeration which should theoretically be produced. The magnitude of this loss, in any given case, is most readily ascertained from thermodynamical charts, but it may also be calculated, in most cases, from the published tables of properties of the various refrigerants in common use. Thus if—

S_1=sensible heat of the liquid approaching the regulating valve,

S_2=sensible heat of the liquid at the temperature of evaporation in the refrigerator,

L_2=latent heat of evaporation at the refrigerator temperature,

then the loss of refrigerating effect expressed as a percentage of that theoretically available is given by—

$$\frac{S_1 - S_2}{L_2} \times 100$$

The following figures indicate the magnitude of this loss for the three principal refrigerants, ammonia, sulphur dioxide, and carbon dioxide, assuming that the liquid refrigerant approaching the regulating valve is at a temperature of 15.5° C. (60° F.), and that the temperature in the refrigerator is − 26.1° C. (−15° F.).

Table II indicates the magnitude of the loss when the refrigerant approaching the regulating valve is at 26.7° C. (80° F.) instead of 15.5° C. (60° F.), the lower temperature remaining at −26.1° C. (−15° F.) as in Table I.

Table I.

	Ammonia	Sulphur Dioxide	Carbon Dioxide
Sensible Heat in Liquid at 15.5°C (60°F) = S_1	30.9	9.3	17.9
Sensible Heat in Liquid at −26.1°C (−15°F) = S_2 ...	−49.4	−14.4	−18.8
Latent Heat of Evaporation at −26.1°C (−15°F) = L_2 ...	583.6	164.2	121.6
% Loss of Refrigerating effect $= \frac{S_1 - S_2}{L_2} \times 100$	13.8%	14.4%	30.2%

How the loss of refrigerating effect varies with temperature conditions, for any one refrigerant, is clearly seen, and also how much better some refrigerants are than others in this respect. It must be clearly understood, however, that the figures given in Tables I and II are not to be taken as indicating the relative merits of the three refrigerating substances mentioned. Thus because Table II shows CO_2 to have a loss of 46 per cent., it is not to say that a CO_2 refrigerating plant is very inefficient. Other factors must be taken into account, and in practice it will be found that CO_2 has much to recommend it. What the figures do tell us, however, is that it is important, if we wish to minimize

this loss, to *cool the liquid refrigerant as thoroughly as possible before it reaches the regulating valve.*

Many ways of doing this have been suggested, some of which are right in principle and some of which are wrong.

The most obvious method is to cool the liquid leaving the condenser, by a further supply of the coldest water obtainable, *e.g.*, from an artesian well. If such a supply is not at hand (it seldom is), then use may be made of the supply of condensing water, although this would not be so effective. Such a method is right in principle, because

Table II.

	Ammonia	Sulphur Dioxide	Carbon Dioxide
Sensible Heat in Liquid at 26.7°C (80°F) = S_1	53.6	16.1	37.2
Sensible Heat in Liquid at −26.1°C (−15°F) = S_2 ...	−49.4	−14.4	−18.8
Latent Heat of Evaporation at −26.1°C (−15°F) = L_2 ...	583.6	164.2	121.6
% Loss of Refrigerating effect $= \frac{S_1 - S_2}{L_2} \times 100$	17.7%	18.6%	46.0%

the heat removed from the liquid refrigerant is carried right away by the cooling water.

Another suggested method is to pass the liquid refrigerant through a coil of pipe immersed in the brine in the refrigerator. The liquid refrigerant would thus be cooled to the refrigerator temperature before going to the regulating valve. Such a system would appear to obviate all the loss indicated in Tables I and II. It is, however, a fallacy and is entirely wrong in principle. A little thought will show that while heat is removed from the liquid refrigerant and taken up by the brine, the brine then simply passes this heat on to the liquid refrigerant

which is evaporating in the refrigerating coils. Further, it will be noted that the heat removed from the liquid refrigerant is not carried away until it has been pumped up from the refrigerator into the condenser. The result then is that there is no increase in the amount of refrigeration produced, and no economy in the power expended, by adopting such a system.

Mention must also here be made of one further system of minimizing the loss at the regulating valve by allowing the expansion of the refrigerant to take place in two stages

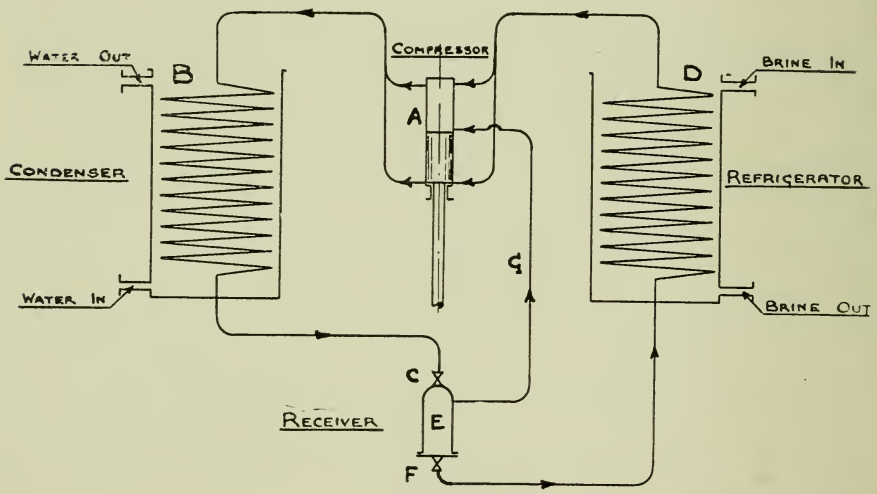

Figure 2

through two regulating valves in succession. The arrangement is shown diagrammatically in Figure 2.

A is the compressor drawing vapour from the refrigerator coils *D*, compressing it and discharging it into the condenser *B*. The liquid refrigerant then goes to the regulating valve *C*, but instead of being expanded into the refrigerator it is expanded into a receiver *E*. This receiver is maintained at some intermediate pressure between that in the condenser and that in the refrigerator. The expansion past the valve *C* causes some of the liquid to evaporate and so cool the remaining liquid, but as the expansion is not so

great as would occur with the ordinary arrangement illustrated in Figure 1, there is less liquid refrigerant evaporated in this way. The vapour formed in the receiver E by such evaporation goes immediately to the compressor by the pipe G, and is at once compressed. The heat removed by it is thus discharged into the condenser without having to be pumped up from the refrigerator. The remaining cooled liquid refrigerant collects in the receiver E and passes through a second regulating valve F into the refrigerator. The refrigerator is thus supplied with liquid refrigerant at a much lower temperature than would ordinarily be the case, and as a result more refrigeration is obtained from the plant. Also since the heat removed from the liquid in the receiver E has not to be pumped up through the whole range of temperature, there is some economy of power per unit of refrigeration. The system is thus sound in principle. The actual results obtained will of course depend upon the working conditions prevailing in each individual case.

Departure of Actual Refrigerating Machine from Ideal.

We have seen that an ideal refrigerating machine would have a coefficient of performance equal to $\frac{T_2}{T_1 - T_2}$, a result not attainable in practice. There are two reasons why our vapour compression machines cannot attain this ideal result. The first is the loss occasioned at the regulating valve. In the ideal machine we should not use a regulating valve, but would allow the expansion to be carried out in an expansion cylinder adiabatically (that is, in a non-conducting cylinder without any gain or loss of heat in the form of heat). If we did this, the expanding liquid refrigerant would do work, helping to drive the plant. In doing this work it would also cool itself and thereby minimize the loss of refrigerating effect which we have already seen occurs at the regulating valve. The use of this ideal expansion cylinder would therefore benefit us in two ways: (1) by giving us greater refrigerating effect, (2) by

reducing the power necessary to drive the machine. In practice, however, it is impossible to have the non-conducting cylinder theoretically necessary; also the expansion cylinder would be so small that the complication it would introduce would outweigh all practical advantages obtainable. The second reason for the non-attainment of the ideal result possible is that the rise of temperature during compression always exceeds T_1, the temperature at which the heat is carried away by the cooling water in the condenser. To limit the rise of temperature to that theoretically necessary, namely, a final temperature of T_1, we should have to work absolutely on the wet compression basis. In many cases to do this would mean carrying such a large quantity of liquid into the compressor that it would be dangerous, and we should reduce the amount of refrigeration obtained from the plant. The importance of the expression $\frac{T_2}{T_1 - T_2}$ is that it shows very clearly that the condenser temperature T_1 and the refrigerator temperature T_2 should be kept as near each other as possible for the least expenditure of power.

Expansion of a Gas in an Expansion Cylinder.

Refrigeration may be produced by the expansion of a gas under conditions which *make it do work*, instead of by evaporating a liquid. In refrigerating work the only gas so used is air, and a diagrammatic sketch of such a machine is given in Figure 3.

A is a compressor which compresses air and discharges it into a cooler *B*, where the heat of compression is removed by cooling water. The compressed and cooled air then passes to an expansion cylinder *C*, in which it does work, helping to drive the compressor. In consequence of the energy given up by it in doing this work, its temperature falls and it leaves the expansion cylinder extremely cold. This cooled air then passes to the refrigerator *D*, where it does useful refrigeration. The air then returns to the compressor for further use. It will be observed that the compression cylinder *A* is the heat pump, pumping the

heat from the refrigerator *D* into the cooler *B*, while the expansion cylinder *C* takes the place of the regulating valve. Machines of this type are now practically obsolete, as they require about five times the power to operate them that a vapour compression machine would take. They do not require any supply of a chemical (NH_3, etc.), however, to enable them to work, and it might easily happen that in a chemical works in a remote situation an air machine would be the only type that could be fitted up for emergency use,

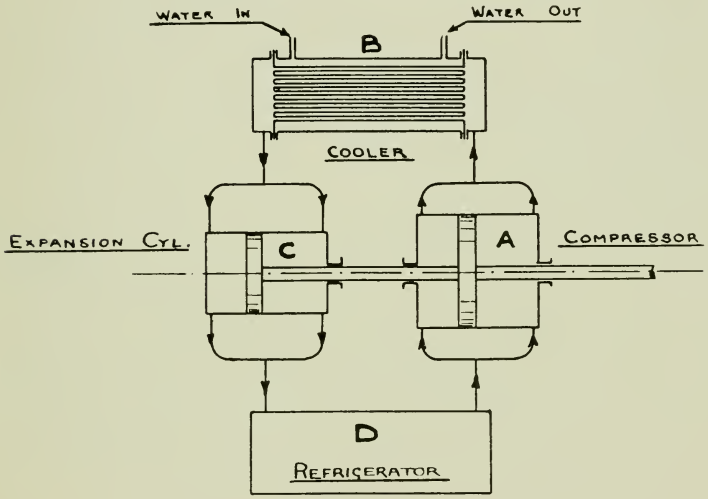

Figure 3

and therefore it is important to understand the principle of their operation.

Absorption System.

If the particular refrigerant used is capable of being readily absorbed by some liquid and as readily given up again, a type of machine known as the *absorption machine* can be constructed and the compressor dispensed with. Such a plant is illustrated in Figure 4.

The vapour coming from the refrigerator coils *D* is taken to a vessel *J* called the absorber. In this the vapour is absorbed in a suitable liquid. The liquor is then removed

by the pump *H* and forced into another vessel *A*, called the generator. In this the liquor is heated and the vapour driven off into the condenser *B* under such a pressure as to bring about liquefaction. The liquid refrigerant then expands through the regulating valve *C* into the refrigerator. In an actual plant several additional pieces of apparatus

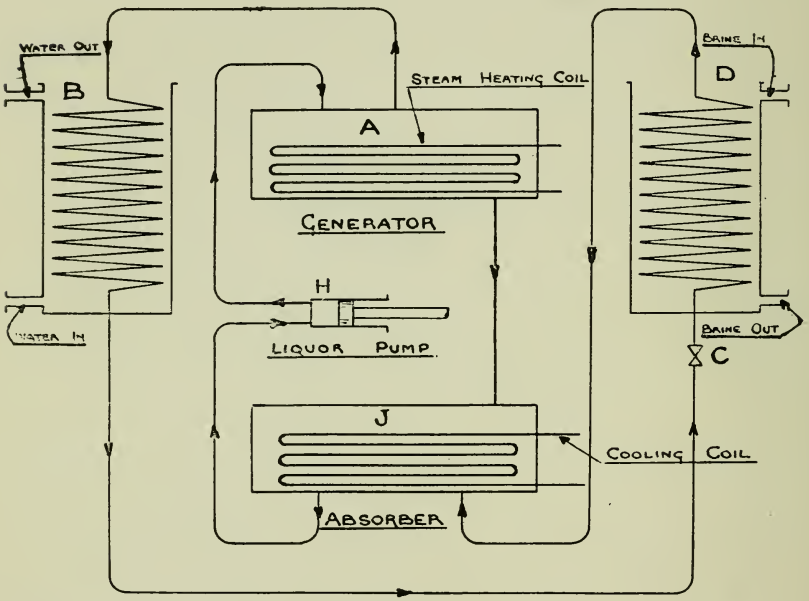

Figure 4

are incorporated to give increased efficiency. The only commercial absorption plants yet evolved are those using ammonia which is absorbed in water. Ammonium nitrate solution has been tried as the absorbing liquid, but such plants have not yet been used commercially. The ammonium nitrate is corrosive, and the vessels in which it is used must be made of or lined with aluminium.

Chapter II.

PROPERTIES OF REFRIGERANTS IN USE.

Introduction.

Many liquids have been suggested for use in refrigerating machines for the production of low temperatures. Theoretically, in an ideal refrigerating machine it does not matter in the least what substance is used. In practice, however, only a few substances are of service to us. Amongst those suggested are ether, ammonia, sulphur dioxide, carbon dioxide, Pictet fluid (this being a mixture of sulphur and carbon dioxides), water, ethyl chloride, and other hydrocarbons. Many points have to be taken into consideration when choosing a refrigerating medium, but most of these points concern chiefly the designer of the machinery, and only indirectly affect the user.

Such points as the user has to consider are intimately bound up with the choice of a machine, and further consideration of the matter is deferred to Chapter IV. It will be sufficient to indicate here some of the chief properties of the more widely used refrigerants.

Ammonia.

The most commonly used refrigerant is anhydrous ammonia. Its chemical properties are well known, and we need not do more than recall its poisonous character if it escapes freely into the atmosphere, especially in a confined space. It also vigorously attacks copper and its alloys. Such metals must not therefore be used in the construction of refrigerating machinery. It is also advisable whenever possible to keep other apparatus in which such metals are used away from any situation where it may come into contact with ammonia leaks. If this cannot be done such apparatus should be protected by suitable fume-tight cases.

Ammonia readily dissolves in the lubricating oil used in

the compressor and may, moreover, cause the oil to saponify. Care must therefore be exercised in the choice of a lubricating oil for ammonia plant.

The pressures met with in ammonia plants are moderate, and do not exceed 200 lb. per square inch, except in tropical situations and other places where the condensing water is warm, say, about 32.2° C. (90° F.).

Liquid ammonia has a large latent heat of evaporation, and therefore each pound of liquid can produce a considerable amount of refrigeration, while the volume of ammonia vapour formed is moderate.

Carbon Dioxide.

Carbon dioxide is the next most widely used refrigerant. It is not poisonous, although it may cause suffocation if present in excess in a confined space. It has no action on metals, and therefore no special precautions have to be taken in this respect. For use in refrigerating machines it should be dry.

The pressures met with in the system may reach 1,200 lb. per square inch or more, and therefore great care has to be taken to make the apparatus strong and to prevent leakage and waste of CO_2. The liquid has not nearly so great a latent heat of evaporation as ammonia, but this is compensated for to a large extent by the very small volume of CO_2 vapour formed from the liquid. Consequently a small compressor can circulate a very large weight of CO_2, and so produce a large amount of refrigeration.

Sulphur Dioxide.

Sulphur dioxide is less widely used than either ammonia or carbon dioxide. The latent heat of the liquid is somewhat greater than that of CO_2, but the volume of vapour produced is large. The outstanding feature of SO_2 is its low pressure, this seldom exceeding 60 lb. per square inch. In fact, in the refrigerator coils themselves the pressure may be lower than atmospheric under some conditions. This is a distinct disadvantage because air may be drawn into the

system and seriously interfere with the working. The air carries in moisture which may freeze and block the pipes on the low temperature side, and the moisture will combine with any traces of SO_3 which may be present, with resultant corrosion of the plant.

Hydro Carbons.

Ether has been given up on account of its inflammability. Ethyl chloride has been used of late as a refrigerating medium, especially for small units such as are supplied to butchers and others.

The pressures met with when using ethyl chloride are very low, and special care must be taken that air is not drawn into the apparatus. The latent heat of evaporation is fairly large, but the volume of ethyl chloride vapour is very large, and for the successful use of ethyl chloride it is necessary to adopt a rotary compressor. Ethyl chloride is inflammable, the extent of its inflammability still being a matter of dispute between its advocates and opponents. A leak of ethyl chloride is harmless to both life and materials of construction. Owing to its solvent action on oils, glycerine must be used as a lubricant for the compressor. It is important to see that ethyl chloride for refrigeration purposes is procured from a reliable source, and does not contain impurities which enable it to dissolve water nor impair its stability. If this is not attended to there may be an action upon the metals used in the construction of the plant. Most ethyl chloride used for commercial refrigeration contains a proportion of methyl chloride, but this does not constitute any disadvantage. Ethyl chloride can also be procured mixed with a quantity of methyl bromide. This renders the mixture non-inflammable, and is said to have no bad effect from the refrigerating standpoint. For refrigerating purposes the substance should be dry.

Ordinarily, ethyl chloride is lighter than the glycerine used for lubrication, but the non-inflammable variety of ethyl chloride is heavier than glycerine.

Such substances as propane, dichlorethylene, trichlorethylene, and carbon tetrachloride have also been suggested as suitable refrigerants. Most of these substances necessitate the use of rotary compressors to deal satisfactorily with the large volumes of vapour formed. No commercial plants using these substances are yet available.

For extra low temperature work, the more common refrigerants or methyl chloride working in conjunction with nitrous oxide or ethylene can be used.

Chapter III.

CALCULATION OF CAPACITY AND POWER FOR A GIVEN MACHINE.

Introduction.

The method of operation of a vapour compression machine has already been explained, and we will consider how to determine the quantity of refrigeration a given machine might be expected to produce.

It will be understood that each pound of refrigerant in circulating round the system will produce a certain quantity of refrigeration. Therefore the total refrigeration produced will depend upon the product: —

$$\left\{\begin{array}{l}\text{Weight of refrigerant circu-}\\ \text{lated per unit time}\end{array}\right\} \times \left\{\begin{array}{l}\text{Refrigerating effect of each}\\ \text{pound of refrigerant.}\end{array}\right.$$

Refrigeration Produced by 1 lb. of Refrigerant.

The refrigerating effect per lb. of refrigerant, it will be remembered, is dependent upon the latent heat of evaporation. We saw in Chapter I that a portion of this latent heat is lost to us during the passage of the liquid refrigerant through the regulating valve. Also a further portion is lost because we carry some of the liquid back, unevaporated, in suspension in the vapour going to the compressor. Deducting these two losses from the latent heat of evaporation, we arrive at the available refrigerating effect of each lb. of refrigerant circulating through the system. In any given case this quantity can be accurately determined from the total heat charts for the substance in question. For our purpose, however, this is not necessary, but the following values may be used instead. The correct value of the refrigerating effect depends upon the temperature and pressure conditions in the condenser and refrigerator coils. With NH_3 and SO_2, however, the variation

under practical working conditions is not serious, and the following figures are safe working values.

Thus the refrigerating effect of 1 lb. ammonia, with the condensing water at 60° F. (15.5° C.), and the temperature of evaporation in the refrigerator at 30° F. (−1.1° C.) would be about 510 B.t.u., while if the refrigerator temperature was reduced to −20° F. (−28.7° C.), the refrigerating effect would be 475 B.t.u. With condensing water at 90° F. (32.2° C.), and refrigerator temperatures as just given, the

Table III.

Refrigerating Effect of 1 lb. CO_2 in B.t.u.

CO_2 Evaporating in Refrigerator at	Condensing Water Temperature.			
	60°F (15.5°C)	70°F (21.1°C)	80°F (26.7°C)	90°F (32.2°C)
30°F (−1.1°C)	80	70	61	54 *
20°F (−6.7°C)	81	72	63	54 *
10°F (−12.2°C)	82	72	63	52 *
0°F (−17.8°C)	82	72	63	48 *
−10°F (−23.3°C) ...	82	72	61	46 *
−20°F (−28.7°C) ...	81	71	58	44 *

*Note.—These values correspond to a condenser pressure of 1,300 lb.

refrigerating effects would be 465 and 410 B.t.u. respectively.

With SO_2 the refrigerating effect per lb. would vary from 146 to 138 B.t.u. with water at 60° F. (15.5° C.), and from 135 to 126 B.t.u. with water at 90° F. (32.2° C.), the refrigerator temperatures being the same as those just quoted for ammonia.

In the case of CO_2 the variation is considerable, and values appropriate to the working conditions should be selected from Table 3.

The temperature at which the liquid refrigerant must be caused to evaporate is generally some 10° to 15° F. (5.5° to 8.3° C.) below the low temperature we wish to produce. Thus if we wanted to cool water to 40° F. (4.4° C.), our refrigerant would be made to evaporate at, say, 25° F. (−4° C.) to 30° F. (−1.1° C.).

Knowing this temperature of evaporation we can find from the tables of properties of the refrigerant we are using, the *volume of vapour which each lb. of the liquid will produce.*

Volume of Refrigerant Circulated by Compressor.

Taking our given compressor, we can calculate from its diameter, stroke, and revolutions, what volume of vapour it will circulate in any given time, assuming it to be perfect and working without loss. If we divide this volume by the volume of 1 lb. of vapour, we know the theoretical weight of refrigerant circulated, and multiplying this by the refrigerating effect per lb., we get the theoretical amount of refrigeration produced.

Loss Due to Clearance.

From this quantity we have to make certain deductions to allow for losses inevitably met with in practice. These losses arise as follows. Firstly, it is necessary in practice to arrange for a small but definite clearance to be left between the end of the piston and the cylinder cover so that there is no risk of knocking the cover off in operation. This space is in general about 1/32nd to 1/16th of an inch. Also pockets occur round the suction and delivery valves, the sum total of such spaces being known as the "clearance volume." This volume remains filled with compressed vapour at the end of each stroke, and until this has expanded again the compressor will not take in any fresh vapour from the refrigerator. Consequently a portion of the movement of the compressor piston, varying up to, say, 25 per cent., is lost to us.

The clearance volume in ammonia machines varies from

$2\frac{1}{2}$ per cent. to 5 per cent. of the volume swept out by the piston. The loss this occasions us may be calculated from the formula: —

$$\text{Loss of capacity} = c \left\{ \left(\frac{p_1}{p_2} \right)^{\frac{1}{n}} - 1 \right\}$$

where c = clearance volume expressed as a fraction of the compressor volume.

p_1 = absolute pressure in cylinder at end of compression.

p_2 = absolute pressure in cylinder at beginning of compression.

n = lies between 1.2 and 1.3.

As a general rule we find that p_1 is approximately the pressure corresponding to a temperature equal to—

condensing water temperature + 10° to 15° F. (5.5° to 8.3° C.).

and is found from the tables of properties of the refrigerant under consideration.

The clearance in CO_2 machines is rather more than in ammonia machines, and may reach $7\frac{1}{2}$ per cent.

Loss Due to Throttling.

A second source of loss is occasioned by the restriction to the free flow of vapour from the refrigerator into the compressor. This restriction may be caused partly by too small connecting pipes, by too small suction valves, and by using too stiff springs on the suction valves. This restriction to free flow increases the volume occupied by each lb. of the vapour, and consequently a compressor of given size cannot circulate so great a weight of refrigerant. This will be clear if we consider the following concrete example. Suppose we have ammonia evaporating at a temperature of, say, 10° F. (−12.2° C.). The pressure in the refrigerator will then be 38 lb. per square inch absolute, and each cubic foot of vapour coming from the refrigerator will weigh 0.1363 lb. If the restriction to the flow causes a loss in pressure between the refrigerator and compressor of, say,

3 lb. per square inch, then our compressor will only be filled with vapour at a pressure of 38−3=35 lb. per square inch. Under these conditions each cubic foot of vapour in the compressor will only weigh 0.1260 lb. Consequently there is a loss in weight circulated equal to 0.1363−0.1260= 0.0103 lb. per cubic foot, or $\frac{0.0103}{0.1363} \times 100$=about $7\frac{1}{2}$ per cent. of the maximum theoretical capacity.

Loss Due to Heat Leakage.

Thirdly, there is a loss caused by heat leakage into the suction pipe and cold parts of the machine. Such leakage not only has to be removed by the machine, together with the useful refrigeration done, but it also causes the vapour going to the compressor to expand. Consequently each cubic foot of vapour weighs less, the compressor handles a less weight of refrigerant than it would otherwise do, and is therefore so much the less capable of producing refrigeration.

Heat leakage, equal to 1 per cent. of the useful refrigeration produced, will cause a reduction of capacity of 0.7 per cent. for SO_2, 1.8 per cent. for NH_3, and from 0.5 per cent. to 2.0 per cent. for CO_2, the latter depending on the temperature conditions. The amount of heat leaking into the cold parts of the machine has never been determined, but probably amounts to 1 or 2 per cent. of the useful refrigeration in large machines to 15 per cent. or more in small ones.

From the theoretical amount of refrigeration which we calculated the machine would produce, we have therefore to deduct

(1) Loss caused by clearance volume.
(2) Loss caused by throttling.
(3) Loss of capacity due to heat leakage.
(4) Dead loss equal to the amount of heat leakage.

The balance will then represent the amount of useful refrigeration which we may expect to get from the machine.

In the distribution of the cold there would, of course, be further losses, due to heat leakage into the insulated

vessels and pipes carrying the cold brine, etc., but these we do not consider at the moment.

Example.

An example will make the method clear. Suppose we have an ammonia compressor with one cylinder 12 inches diameter, 24-inch stroke, double acting, running at 70 r.p.m., and with a compressor rod 3 inches diameter. What amount of refrigeration would it produce if it was cooling brine to 20° F. (−6.7° C.), the condensing water being at an initial temperature of 70° F. (21.1° C.)?

Area of compressor piston = 113 square inches.
Area of compressor rod = 7 square inches.
Mean area of both sides of piston = 109.5 square inches.
Compressor displacement per revolution = 3.04 cubic feet.
Compressor displacement per hour = $3.04 \times 70 \times 60 = 12{,}760$ cubic feet.
Temperature to which brine is cooled = 20° F. (−6.7° C.).
Temperature at which ammonia evaporates = 20 − 10 = 10°F. (−12.2° C.).
Pressure under which ammonia evaporates = 38 lb. square inch abs.
Weight of 1 cubic foot ammonia vapour under these conditions = 0.1363 lb.
Theoretical weight of ammonia circulated = $12760 \times 0.1363 = 1740$ lb. per hour.
Available refrigerating effect of each lb. ammonia, say, 490 B.t.u.
Theoretical refrigeration produced = $1740 \times 490 = 852000$ B.t.u. per hour.

Losses in Given Example.

From our previous rule we find the condenser pressure to be about 154 lb. per square inch, and assuming a clearance of 5 per cent., or c = 0.05 the loss due to clearance is—

$$\text{Loss} = 0.05 \left\{ \left(\frac{154}{35} \right)^{\frac{1}{1.3}} - 1 \right\} = 10.6 \text{ per cent.}$$

Again assuming a throttling of 3 lb. per square inch, our compressor is only filled with ammonia at 38 − 3 = 35 lb. pressure.
Weight of 1 cubic foot vapour at 38 lb. pressure = 0.1363 lb.
Weight of 1 cubic foot vapour at 35 lb. pressure = 0.1260 lb.

Loss due to throttling $= \frac{0.1363 - 0.1260}{0.1363} \times 100 = 7\frac{1}{2}$ per cent.

Assuming a heat leakage of 5 per cent. for a machine of the size in question, we have a further loss of capacity of $5 \times 1.8 = 9$ per cent.

Our total losses are therefore—

	Per cent.
(1) Clearance	10.6
(2) Throttling	7.5
(3) Loss of capacity due to heat leakage ...	9.0
(4) Dead loss equal to amount of heat leakage ...	5.0
Total	32.1

Balance—useful refrigeration $= 100 - 32 = 68$ per cent.

Therefore the amount of useful refrigeration we could expect is—

$852{,}000 \times 0.68 = 580{,}000$ B.t.u. per hour.

It will be noted that we assume the machine to be in good order, with no leakages past the piston and valves. Any such leakages will, of course, further reduce the amount of refrigeration produced.

We further assume that the quantity of piping in the condenser and refrigerator is sufficient for the amount of work to be done. If either of these parts of the plant are too small then the calculated capacity of the plant will not be realised.

Whatever may be the refrigerant used, the method of calculation is the same in all cases, substituting, of course, the figures for temperatures, pressures, etc., corresponding to the conditions of the case and the particular refrigerant used.

The foregoing example has been worked out in detail in order to show clearly the method involved and the magnitude of the respective items of loss. To assist in making rapid approximate calculations in practice, Tables 4, 5, and 6 have been compiled, which state the theoretical number of B.t.u. per hour which each cubic foot of compressor displacement can produce.

To obtain the theoretical amount of refrigeration which any given machine will produce, multiply the compressor displacement in cubic feet by the figure selected from the tables corresponding to the proposed temperature condi-

Table IV.

AMMONIA REFRIGERATING MACHINES.

Theoretical B.t.u. of refrigeration which one cubic foot of compressor displacement will produce under the given temperature conditions.

TEMPERATURE AND PRESSURE UNDER WHICH LIQUID AMMONIA EVAPORATES.	CONDENSING WATER TEMPERATURE AND CONDENSER PRESSURE			
	60°F 129 lb.	70°F 153 lb.	80°F 182 lb.	90°F 214 lb.
30°F 59.8 lb.	106	103	102	96
20°F 47.7 lb.	86	84	81	77
10°F 38 lb.	69	67	64	61
0°F 29 lb.	55	52	50	47
−10°F 23.3 lb.	42	40	38	36
−20°F 17.9 lb.	32	31	29	28

tions. From this quantity deduction must be made for the losses previously mentioned. Figure 5 shows graphically the losses due to clearance volume, with varying clearances and ratios of compression. Tables 7 and 8 give the losses due to throttling for ammonia and sulphur dioxide. The

loss due to throttling with CO_2 is practically negligible, being only 2.3 per cent. for the worst conditions given in Tables 7 and 8. The losses due to heat leakage can be calculated from the data already given.

Table V.

CARBON DIOXIDE REFRIGERATING MACHINES.

Theoretical B.t.u. of refrigeration which one cubic foot of compressor displacement will produce under the given temperature conditions.

TEMPERATURE AND PRESSURE UNDER WHICH LIQUID CARBON DIOXIDE EVAPORATES.	CONDENSING WATER TEMPERATURE AND CONDENSER PRESSURE.			
	60°F 848 lb.	70°F 962 lb.	80°F 1100 lb.	90°F 1300 lb.
30°F 487 lb.	463	405	353	313
20°F 422 lb.	393	349	306	262
10°F 362 lb.	335	294	257	212
0°F 308 lb.	281	247	216	168
−10°F 260 lb.	213	188	159	120
−20°F 218 lb.	195	171	140	106

These tables also make it clear how the quantity of refrigeration produced by a given machine varies with the temperature conditions; the lower the refrigerator temperature and/or the higher the condenser temperature, the less the amount of refrigeration produced.

Power Required to Drive Machine.

In addition to knowing how much refrigeration a given machine will produce, we also require to know the power necessary to drive it. This can be readily obtained from Tables 9, 10, and 11, which give the average pressure in the compressor cylinder under various conditions of

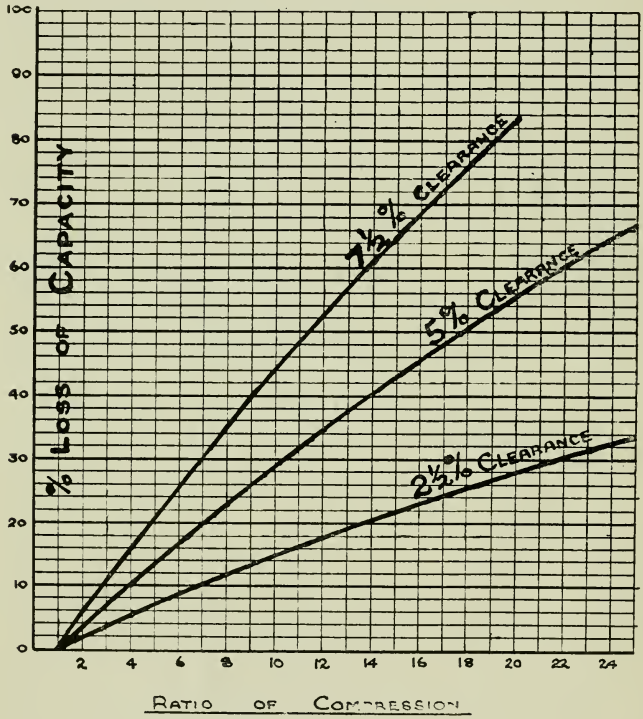

Figure 5

working. The power required is then given by the usual formula—

$$\frac{\text{average pressure} \times \text{area of piston} \times \text{stroke} \times \text{effective strokes per minute}}{33{,}000}$$

This power should be divided by a figure varying from 0.75 to 0.85 to allow for the energy spent in overcoming

the friction of the machine, the result giving the horse-power to be applied at the compressor shaft or belt pulley.

A study of these mean pressure tables is interesting. They show that for any given condensing water temperature there is a refrigerator temperature which requires a maximum

Table VI.

SULPHUR DIOXIDE REFRIGERATING MACHINES.

Theoretical B.t.u. of refrigeration which one cubic foot of compressor displacement will produce under the given temperature conditions.

TEMPERATURE AND PRESSURE UNDER WHICH LIQUID SULPHUR DIOXIDE EVAPORATES.	CONDENSING WATER TEMPERATURE AND CONDENSER PRESSURE.			
	60°F 49 lb.	70°F 59 lb.	80°F 71 lb.	90°F 99 lb.
30°F 21.5 lb.	41	40	39	38
20°F 17 lb.	33	32	31	30
10°F 13.3 lb.	26	25	24	24
0°F 10.4 lb.	20	19	19	18
−10°F 7.9 lb.	15	15	15	14
−20°F 5.9 lb.	12	11	11	11

horse-power, while for other refrigerator temperatures above or below this, the power required is less. The point of maximum horse-power also varies with the water temperature. When installing a plant therefore which may at times have to work over a range of temperatures the prime mover must have a sufficient margin of power to work over this maximum point.

Table VII.

AMMONIA REFRIGERATING MACHINES.

Per cent. loss of capacity due to throttling at the compressor inlet.

TEMPERATURE UNDER WHICH LIQUID AMMONIA EVAPORATES.	AMOUNT OF THROTTLING LB. PER SQ. IN.				
	1	2	3	4	5
30°F	0.9	2.5	4.1	5.8	7.4
20°F	1.8	3.9	5.8	7.8	9.8
10°F	2.6	5.1	7.6	10.2	12.6
0°F	3.3	6.2	9.5	12.8	15.7
−10°F...	4.2	8.0	12.0	16.3	20.2
−20°F...	5.2	10.6	16.0	21.3	26.7

Table VIII.

SULPHUR DIOXIDE REFRIGERATING MACHINES.

Per cent. loss of capacity due to throttling at the compressor inlet.

TEMPERATURE UNDER WHICH LIQUID SO_2 EVAPORATES.	AMOUNT OF THROTTLING LB. PER SQ. IN.				
	1	2	3	4	5
30°F	4.4	8.5	13.2	17.3	22.0
20°F	5.2	10.6	16.0	22.0	28.1
10°F	7.0	14.0	20.7	28.5	36.3
0°F	8.5	18.0	25.9	34.9	45.7
−10°F...	11.5	24.5	—	—	—
−20°F...	—	—	—	—	—

Table IX.

AMMONIA REFRIGERATING MACHINES.

Mean pressure in the compressor cylinder under given working conditions.

TEMPERATURE AND PRESSURE UNDER WHICH LIQUID AMMONIA EVAPORATES.	CONDENSING WATER TEMPERATURE AND CONDENSER PRESSURE.			
	60°F 129 lb.	70°F 153 lb.	80°F 182 lb.	90°F 214 lb.
30°F 59.8 lb.	50.0	62.0	75.0	87.0
20°F 47.7 lb.	52.0	63.0	73.0	84.0
10°F 38 lb.	52.0	61.0	70.0	79.0
0°F 29 lb.	51.0	58.0	65.0	72.0
−10°F 23.3 lb.	47.0	53.0	59.0	65.0
−20°F 17.9 lb.	44.0	49.0	54.0	59.0

Table X.

CARBON DIOXIDE REFRIGERATING MACHINES.

Mean pressure in the compressor cylinder under given working conditions.

TEMPERATURE AND PRESSURE UNDER WHICH LIQUID CARBON DIOXIDE EVAPORATES.	CONDENSING WATER TEMPERATURE AND CONDENSER PRESSURE.			
	60°F 848 lb.	70°F 962 lb.	80°F 1100 lb.	90°F 1300 lb.
30°F 487 lb.	287	357	435	538
20°F 422 lb.	320	383	453	543
10°F 362 lb.	342	400	461	532
0°F 308 lb.	354	405	457	507
−10°F 260 lb.	355	401	441	474
−20°F 218 lb.	353	391	412	435

Table XI.

Sulphur Dioxide Refrigerating Machines.

Mean pressure in the compressor cylinder under given working conditions.

Temperature and Pressure under which Liquid Sulphur Dioxide Evaporates.	Condensing Water Temperature and Condenser Pressure.			
	60°F 49 lb.	70°F 59 lb.	80°F 71 lb.	90°F 99 lb.
30°F 21.5 lb.	19.6	24.6	29.7	38.4
20°F 17 lb.	20.4	24.6	28.9	36.9
10°F 13.3 lb.	20.2	23.7	27.2	34.0
0°F 10.4 lb.	19.2	22.2	25.0	30.8
−10°F 7.9 lb.	17.7	20.2	22.3	27.3
−20°F 5.9 lb.	16.1	17.8	19.7	23.5

Chapter IV.

CALLING FOR TENDERS AND CHOICE OF A SYSTEM.

Introduction.

The foregoing chapters will have made it clear that the amount of refrigeration produced by a given machine is a very variable quantity depending upon the working conditions.

Therefore the *size* of any plant can only be *definitely specified* if we state both the *amount of refrigeration and the temperature conditions under which it is to be produced.* This, and this alone, forms a definite specification of size.

Statement of Amount of Refrigeration.

When inviting tenders for a plant, the first thing to be specified is the amount of refrigeration it is required to produce. This quantity is an amount of heat, and should therefore be specified in some recognised *heat unit*, such as the *calory* or *British thermal* unit. It is usual in this country to specify the size in B.t.u. per hour.

It should be noted, however, that another unit is sometimes used in the refrigerating industry, known as the " Ton of Refrigeration " One " ton of refrigeration " is considered to be equal to the amount of heat required to melt one ton of ice at 32° F., into water at 32° F., during a period of 24 hours. It is thus dependent on the latent heat of fusion of ice and the magnitude of the ton. Taking the latent heat of fusion of ice to be approximately 144 B.t.u. per lb., the English ton of refrigeration is equal to

$$2{,}240 \text{ lb.} \times 144 = 322{,}000 \text{ B.t.u. per 24 hours.}$$

This, it must be understood, is an approximate value because we have used a round figure for the latent heat of fusion, and it must not be forgotten that such an amount of refrigeration is spread over a period of 24 hours.

In the U.S.A. the ton of refrigeration is only equal to

$$2{,}000 \times 144 = 288{,}000 \text{ B.t.u. per 24 hours.}$$

because the American ton is one of 2,000 lb.

Some refrigerating machine builders in this country have adopted the American ton for the purpose of rating the size of their machines. Since therefore the American unit is smaller than the older English unit, a given machine will have an apparently greater capacity in tons refrigeration if rated with the American unit than if rated with the English unit. Thus a machine which could produce refrigeration equal to 10,000,000 B.t.u. in 24 hours would have the following ratings on the British and American systems respectively: —

$$\text{British unit } \frac{10{,}000{,}000}{322{,}000} = 31.0 \text{ tons refrigeration.}$$

$$\text{American unit } \frac{10{,}000{,}000}{288{,}000} = 34.7 \text{ tons refrigeration.}$$

As tenders do not always make it clear which unit has been used, it is important that intending purchasers should ascertain this point before concluding that one manufacturer is offering a larger machine than another. It is preferable to avoid the "ton of refrigeration" and use a definite heat unit.

Statement of Temperature Conditions.

The second point to be made clear when inviting tenders is the low temperature to be produced. In the application of refrigeration to chemical processes it may be a matter of some difficulty to decide what is the actual low temperature to be specified in the enquiry unless an expert is consulted. The works chemist or engineer will know, of course, what low temperature it is desired to produce in the particular chemical process under consideration, but the temperature which the refrigerating machine will have to produce may be very much lower than that required in the process. This is because it is seldom possible to apply directly to a chemical process the cold produced by the machine. The

following example will make this clear. Suppose we desired to use an ammonia machine to cool moist chlorine. The ammonia must be evaporated in iron pipes, yet iron pipes are not permissible in contact with moist chlorine. We should therefore be obliged to let the ammonia cool, say, calcium chloride brine, and then circulate this cold brine through (or around) lead pipes in contact with the chlorine. If, therefore, we desired to cool such chlorine to, say, 40° F. (4.4° C.), the temperature of brine required to do this would depend largely upon the design and size of the cooler constructed of lead pipes, and would probably be 30° F. (−1.1° C.) or less. The ammonia would have to evaporate at a temperature some 10° to 15° F. (5.5° to 8.3° C.) below that of the brine. There is thus an element of uncertainty introduced in the temperature requirement.

In general, the works staff would have to undertake the design and construction of such apparatus as a lead pipe cooler, this being outside the scope of the refrigerating machine contractor.

It must again be emphasized, however, that the lower the temperature at which the refrigerant has to evaporate, the greater the size of machine for a given duty, the more the first cost, and the greater the operating cost.

If an intermediate agent such as brine must be employed, the enquiry should state the temperature to which the brine is to be cooled. If the cold is being applied direct, then the actual temperature to be produced in the process should be specified.

The next points to be specified are the temperature of the water available for the condenser, the quantity of water available, whether plentiful or scarce, and the nature of such water supply, whether fresh, salt, corrosive, dirty, etc. These points influence the size and type of condenser which should be installed.

Standard Temperature Conditions.

It has been suggested that arbitrary standard working temperature conditions should be adopted by the refrigera-

ting industry under which to rate or state the size of a machine, that is, the amount of refrigeration it will produce. The idea of such standards is to eliminate the effect of varying temperature conditions and so make a direct comparison possible between the proposals of different manufacturers. At present we have some English machine builders with standards of their own, different from those of any other maker. Consequently any statement of size or rating on one maker's standard is not comparable with a similar statement by another maker. Definite standards have been adopted in the U.S.A., who have fixed the working conditions as follows: —

The outlet pressure (from the compressor) to be that corresponding to a saturation temperature of 86° F. (30° C.).

The inlet pressure (to the compressor) to be that corresponding to a saturation temperature of 5° F. (−15° C.).

Apart from any technical objections which can be lodged against such a method of specifying standard conditions of working, it should be particularly noted that no statement of capacity or size of a machine under any arbitrary standard conditions, whatever they are, is of much value to the purchaser of a refrigerating plant. The capacity under actual conditions of working will be very different from that under standard conditions, and further two machines which will do equal work under standard conditions will not necessarily do equal work under any other conditions. Therefore the only safe course for the purchaser is to insist on a statement of the amount of refrigeration which will be produced under the actual conditions which the machine must work under when installed.

Standard working conditions are of more value for scientific than for commercial purposes.

General Information.

The enquiry must also state the nature of the power supply available for driving the plant.

Where possible, some information as to the nature of the process or the substance to be cooled and the method of applying the cold should be given. If such information must, however, be kept secret, then it will only be possible to specify the amount of refrigeration to be produced and the working temperature conditions. The works staff would in such a case have to assume the responsibility for the production of the desired result in the chemical process.

Information to be Given with Tender.

The machine builder should also be instructed to state, in the tender, the following information: —

(*a*) Diameter, stroke, and revolutions per minute of the machine offered for the stated duty.

(*b*) Power required to drive the machine under working conditions stated.

(*c*) Amount of pipe surface in the condenser.

(*d*) Amount of pipe surface in the refrigerator.

(*e*) Quantity of water required for the condenser.

From such data it is possible to make a fair comparison between the various offers received.

Contractor's Guarantee.

Apart from the usual guarantees for materials and workmanship, a guarantee is frequently given that the machine offered is capable of producing the refrigeration required under the stated working conditions. The usual penalty for failure to fulfil this guarantee is the removal of the plant and the refunding of all moneys paid on account of the same.

It should be noted, however, that such a guarantee clause does not necessarily form any protection to the purchaser unless the design of the plant permits of the capacity being satisfactorily tested after installation in the purchaser's works. For a satisfactory test it is necessary that the refrigeration produced can be readily measured. This is only the case where the plant is cooling a liquid, the rate

of circulation of such liquid and its fall of temperature being measured. Similar measurements, but with less accuracy, can be made when cooling an innocuous gas such as air. In most other cases indirect measurements only are possible, and will not necessarily be accepted by the contractors. A further difficulty in arranging for a satisfactory test is caused by the, generally, small change of temperature of the fluid being cooled. Thus, in the case of a brine cooler, the fall of temperature of the brine in passing through the cooler may be less than 5° F. (2.8° C.). Consequently a small error in temperature measurement will make a large difference in the amount of refrigeration apparently produced.

Tests may be arranged at the makers' works, the machine being made to cool a liquid for the purpose of the test. Such a test would not prove the efficiency of a set of refrigerator coils designed for cooling a gas, as these could not be used for cooling the liquid for test purposes.

It is therefore important to remember when considering any such guarantee of capacity, whether a test can be made after installation and the capacity directly measured. If not, then there may be considerable difficulty in establishing whether such guarantee has been fulfilled or not.

Again, supposing it is established that the guarantee has not been fulfilled, and the maker removes the plant and refunds all payments made to him, the position is still unsatisfactory from the purchaser's point of view. He must begin over again the installation of a fresh plant with a considerable loss of time in getting the process working and incurs a consequent loss of revenue. It is therefore preferable to be quite certain that an installation is going to be satisfactory and not place reliance on guarantees.

Choice of a Refrigerant.

We have indicated in Chapter 2 some properties of the various refrigerants in common use. Which of these, if any, should be adopted in preference to the others? It cannot be said that any one is the best for our purpose.

The first consideration must be "What is the low temperature required?" It is most inadvisable that the pressure under which the refrigerant evaporates should be too low because the machine then becomes bulky for the work done,

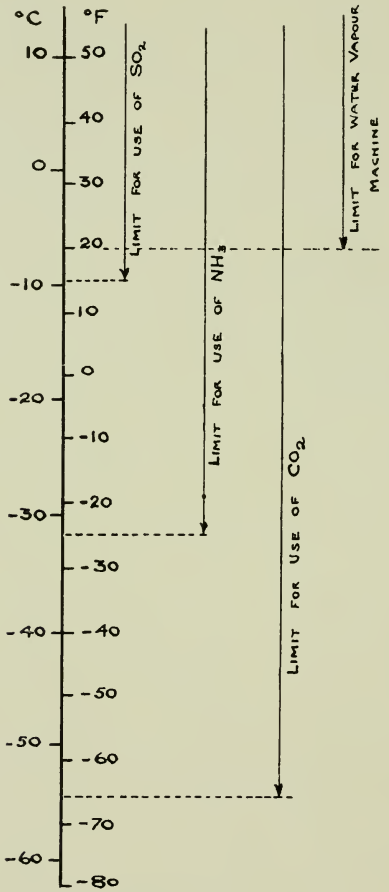

Figure 6

and consequently less efficient and more costly. Also with the reciprocating type of compressor, the refrigerator pressure should never be below atmospheric pressure, so

that we may avoid trouble due to air and moisture entering the system. On this basis we can construct Figure 6, which indicates the limits of operation with the various refrigerants.

Under such a criterion the limit to the use of SO_2 would arise at a temperature of 15° F. (−9.4° C.), the limit for NH_3 would be −25° F. (−31.8° C.), while with CO_2 considerably lower temperatures can be reached. The author has in fact produced temperatures between −60° F. (−51° C.). and −70° F. (−56.7° C.) on a large scale in chemical operations with CO_2 machines.

Other factors to be considered are reliability, first cost, and operating cost.

As regards reliability well designed CO_2 or NH_3 machines are equally reliable. Small CO_2 machines with indifferent attention and intermittent working are liable to need greater replenishment of their charge than ammonia machines because CO_2 is practically inodourous and small but serious leaks can pass unnoticed. No leak of ammonia can escape attention.

There is not a great deal of difference in either the first cost or the running cost of moderate size ammonia and carbon dioxide machines. In large plants both the first cost and the running cost will be less with ammonia than with CO_2. Undoubtedly CO_2 requires more power per unit of refrigeration but, in moderate size plants, this is masked by the power required to overcome friction and to work the auxiliaries. SO_2 plants are now out of favour except in very small sizes.

An advantage of CO_2 is its ability to work with copper condensing coils. This is a matter of importance where the condensing water is actively corrosive to iron.

In general the author considers that an ammonia plant will be found the best all round proposition. Exceptions to this statement are cases in which—

(1) a very low temperature is required, say, where the refrigerant would have to evaporate at or below −25° F. (−31.8° C.).

(2) where any escape of ammonia might constitute a serious danger—conditions seldom, if ever, arising in chemical works.

(3) where condensing water is actively corrosive.

(4) where ammonia is difficult to get or where CO_2 can be obtained or is manufactured on the works.

(5) where it is advantageous to adopt a combined solvent and refrigerating process, when a hydro-carbon refrigerant may be used.

Chapter V.

CHOICE OF A MACHINE.

Slow Speed and High Speed Machines.

For use on land, refrigerating machines can be conveniently divided into two classes:—

(*a*) slow speed.
(*b*) high speed.

In such a classification the term "speed" refers more particularly to revolutions and not piston speed. With the slow speed type, the rate of revolution would vary from, say, 40 per minute in very large units, to, say, 150 per minute in small units; while with the high speed type the revolutions might be anything up to 550 per minute, depending on the particular design of machine.

The outstanding features of the slow speed machine are its reliability, its accessibility, and its small wear and tear. Unless, however, it is directly driven by a slow speed engine, some form of power transmission, belts, gears, etc., is necessary. The total floor space taken up by a slow speed unit and its prime mover is large.

The introduction of the high speed unit was met with the objections that it was less accessible, less reliable, more dangerous, and the wear and tear would be excessive. Considering these objections quoted, certainly the high speed machine is not so accessible as the slow speed because the high speed machine is invariably built with the moving parts totally enclosed. Such a construction is, however, so common in many classes of machinery at the present day that no skilled mechanic should have any difficulty in dealing with the enclosed construction, and the objection cannot be considered valid. With regard to reliability and wear and tear, this is a matter that experience only can decide. The author's experience with several high speed machines of different makers has not indicated any lack of

reliability or more wear and tear over the slow speed machine. It is, of course, understood that the high speed machines are well designed and built by makers of repute. With regard to the high speed machine being more dangerous, this idea has arisen because we generally carry into the compressor a small quantity of liquid refrigerant to prevent overheating. On occasions an excess of such liquid may enter the cylinder, and it was considered that the high speed of revolution did not give sufficient time for such liquid to be forced out through the delivery valves without setting up undue forces or knocking, and a consequent risk of damage. The high speed machines under the personal supervision of the author have been found to clear themselves of any excess liquid which may be met with, without trouble. This applies to both single acting and double acting high speed machines. Therefore none of the objections usually lodged against a high speed machine are valid, providing the design and manufacture are right. The economy of space which such high speed machines permit is enormous, while the possibility of direct coupling them to their prime movers permits of a reduction in first cost and in running costs.

Thus, for example, a double cylinder double acting vertical high speed unit direct coupled to an electric motor, and having a compressor displacement of 520 cubic feet per minute, could be accommodated in a floor space of 5 feet by 10 feet. A horizontal steam driven unit of 400 cubic feet compressor displacement would require a floor space of some 13 feet by 22 feet, not counting the space occupied by the boiler plant. A similar size slow speed horizontal machine belted to an electric motor would require floor space of a maximum width of some 10 feet by a maximum length of some 42 feet.

Slow Speed Horizontal Units.

Figure 7 shows a typical horizontal slow running ammonia machine driven by belt, while Figure 8 gives a section through a compressor cylinder.

Machines by various manufacturers follow very closely the arrangement illustrated in Figures 7 and 8, although, of course, there are differences in detail. The outstanding features are the spherical cylinder covers, which are used so that large valves can be fitted; the spherical ended piston, which is shaped to fit the covers and reduce the clearance

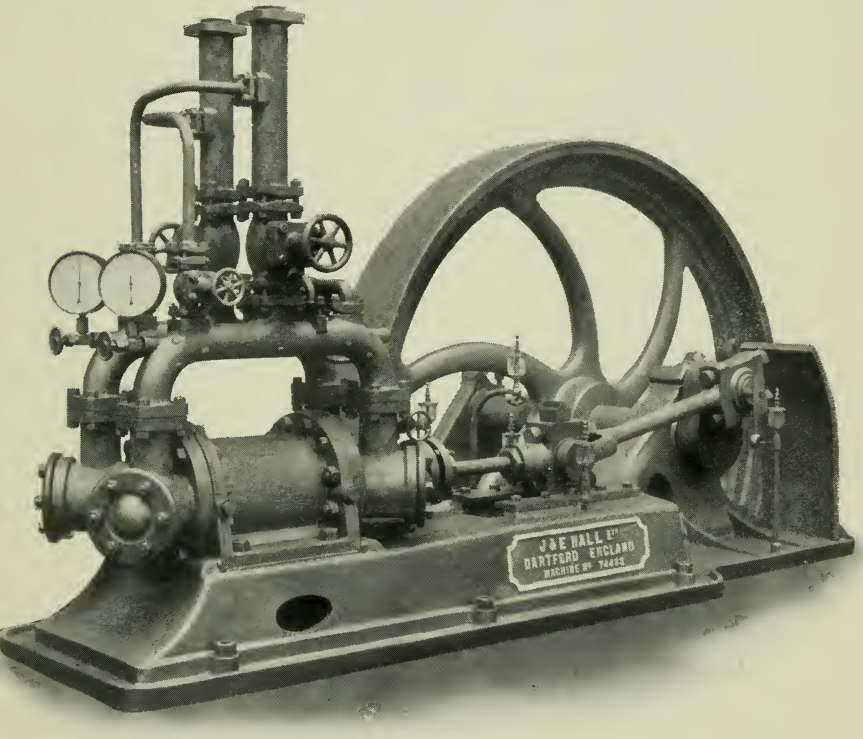

Figure 7

volume to a minimum; and the long stuffing box for the compressor rod. The packing in such stuffing boxes is divided into two parts by a metal ring called a lantern. The space which the lantern creates in the stuffing box is connected by a small pipe to the compressor suction, and so reduces the chances of ammonia escaping into the atmos-

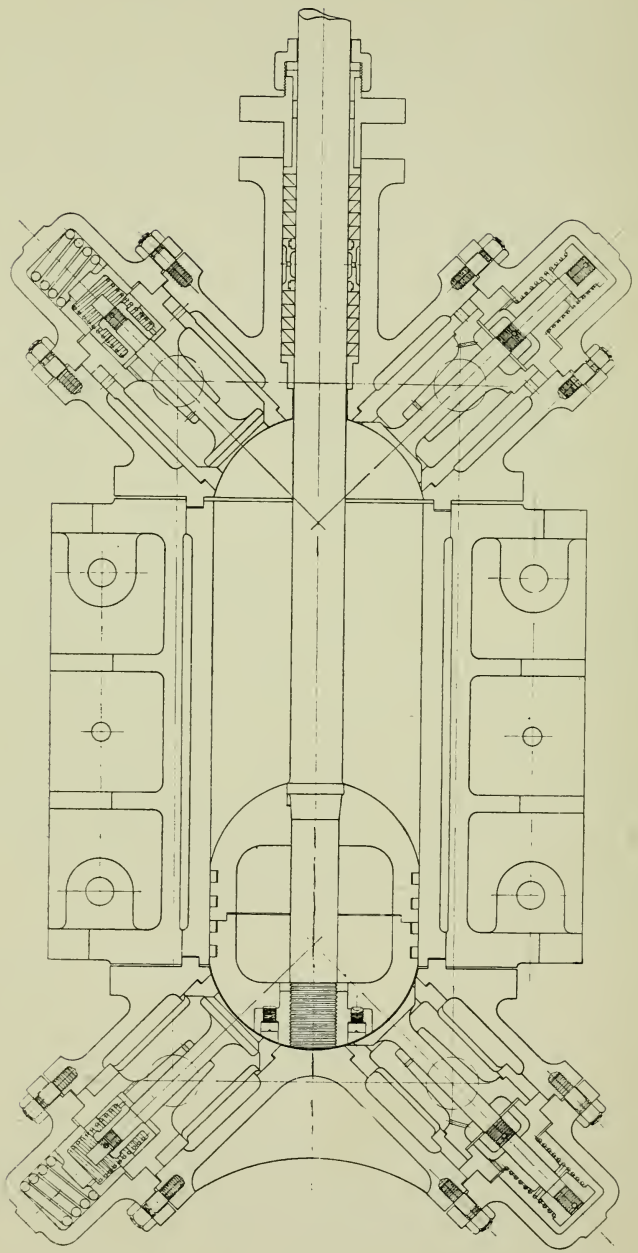

Figure 8

phere. The stuffing boxes are now frequently packed with a metallic packing. It should also be noted that with the type of compressor shown the suction valves project into the cylinder when open. Means must therefore be provided, in the design, to prevent the head of the valve falling into the cylinder in the event of the valve spindle breaking.

In all cases a certain scheme of pipe connections and stop valves at the compressor should be insisted on. Such a scheme is shown diagrammatically in Figure 9.

Thus, 1 is the main discharge stop valve, isolating the compressor from the condenser. This is frequently of the

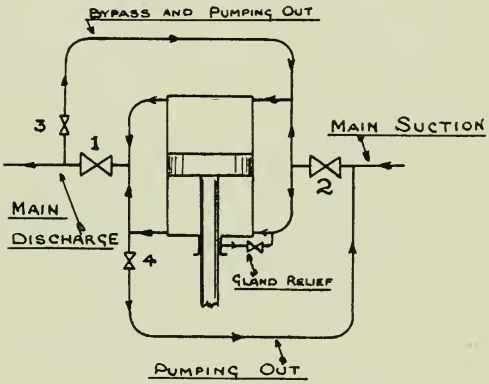

Figure 9

non-return type, or alternatively, an additional non-return valve is fitted. The main suction stop valve is shown at 2. Additional connections of smaller bore than the main pipes, and carrying valves 3 and 4, are arranged as shown. The purpose of these valves is twofold. If valves 1 and 2 are closed while 3 and 4 are open, the compressor will withdraw gas from the condenser and discharge it into the refrigerator, thus making the circulation exactly the reverse of that in normal working. Operation in this manner is carried on when we want to empty the condenser for repairs, and do not want to throw away the refrigerant stored up in the condenser. Again, at starting, it is desirable, especially if the prime mover is an internal combustion engine, to relieve

the load until the machine has got fairly moving. We should therefore start with the main suction stop valve (2) closed, the main discharge valve (1) and the auxiliary valve (3) would be open. Auxiliary valve (4) would be closed and not touched in normal running. The opening of valve (3) puts both ends of the compressor (suction and delivery) into direct communication with the condenser, and therefore at the same pressure. Consequently the compressor piston is balanced, and the machine starts without load. When up to speed, valve (3) is gradually closed, causing the compressor to begin to circulate. As soon as (3) is closed, the main suction stop valve (2) is gradually opened, and the compressor takes up its full work.

All machines should be fitted with an oil separator on the discharge side of the compressor to remove lubricating oil carried over with the ammonia. As a rule, such separation is effected by reducing the velocity of the gas, causing the oil to drop out of suspension, or by centrifugal action. Neither type is completely effective, especially if the compressor discharge is hot. This has caused one manufacturer to introduce a water-cooled separator to eliminate the oil better. In any case, it would be an improvement to fit the usual oil separator close up to the condenser, instead of close to the compressor. It would then have to deal with as cool gas as possible. To remove the final traces of oil, it is advisable in ammonia plants to fit a vessel at the exit from the condenser in which the liquid ammonia can collect. Such a vessel is sometimes called a "pig." The oil, being of greater density than the liquid ammonia, collects at the bottom of the vessel, and after the plant has been standing, say, overnight, or over a week-end, there is a sharp demarcation between the two, thus permitting the oil to be easily withdrawn. In order to avoid the loss of the ammonia dissolved in oil from the separators, etc., it is usually discharged into another vessel, called a rectifier. The rectifier can be connected to the compressor suction, and any ammonia coming from the oil pumped away. Figure 10 shows diagrammatically the connections for the oil separator and rectifier.

It is becoming more usual, but is not yet general practice, to fit safety valves on ammonia plant, to provide against excessive pressure in any part of the system. The frequent absence of this obvious safeguard is due to the difficulty of preventing such a device leaking after it has once operated.

SO_2 compressors often follow the same general design just shown for ammonia compressors, and therefore do not need special illustration.

The construction of CO_2 machines is entirely different from that of ammonia machines, owing to the high pressures

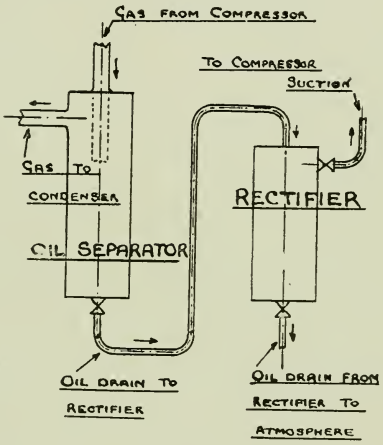

Figure 10

met with. It is usual to design CO_2 machinery for a maximum pressure of 1,500 lb. per square inch. For such pressures the compressor cylinder is made out of a solid steel forging, a typical machine being shown in section in Figure 11. The long stuffing box and the lantern will be noticed again. Instead, however, of this lantern space being connected to the suction, as in the case of ammonia machines, it is connected to an oil pump which automatically pumps oil into the lantern space at a pressure higher than that of the CO_2 in the condenser. If there is any leakage, it is therefore of oil and not of CO_2. The stuffing box is packed either with U rings as shown, or metallic packing

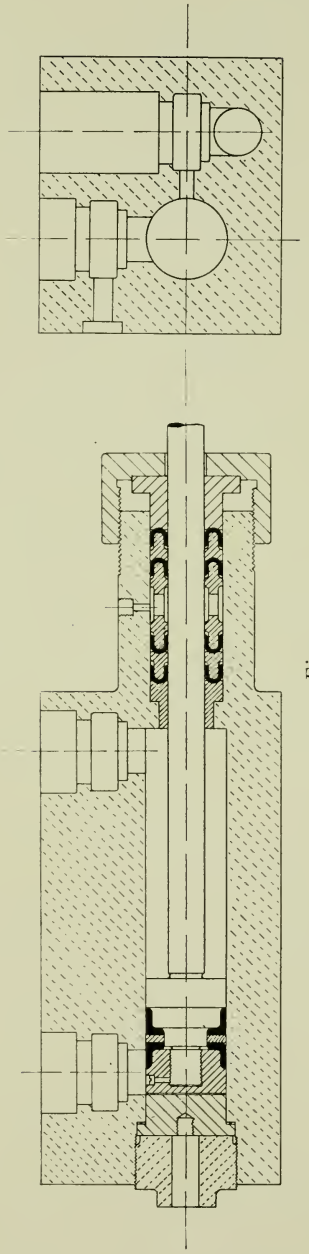

Figure 11

may be used. The compressor pistons were formerly always fitted with leather packing rings as shown in Figure 11, but now metal piston rings are mostly used. Metal rings are much to be preferred as they are not damaged if the compressor should become overheated. CO_2 machines also are fitted with oil separators, but we do not fit storage vessels for the liquid CO_2, nor rectifiers. It is also not usual to fit the cross connections shown in Figure 9 to enable the CO_2 to be pumped out of the condenser. This is because CO_2 is so cheap, compared with other refrigerants. The gas is simply blown out of any part of the plant requiring repair.

The general arrangement of a horizontal slow speed CO_2 compressor does not differ from the arrangement of the ammonia or SO_2 machine, except in having a special design of cylinder.

High Speed Units.

Modern high speed machines are of two distinct types:

(1) Single acting vertical machines with trunk pistons.

(2) Double acting machines either vertical or horizontal.

Figure 12 shows a machine of the first type. It will be noticed that the trunk piston is attached directly to the connecting rod, there being no compressor rod and stuffing box. The suction and delivery valves are carried in the cylinder cover as in the horizontal machines.

Figure 12a shows a larger and more modern type of high speed single acting machine with valves in the cylinder cover. The valves here are of a special light-weight pattern, and not poppet valves as in Figure 12. The machine is arranged for direct coupling to a motor, but can also be belt driven.

Figure 13 shows another variety of the single acting vertical type, in which the suction valves are mounted in the piston itself, the ammonia entering the compressor passing upwards through the piston. The delivery valves are located in what is really a false cylinder cover, this false cover being held down by springs from the ordinary

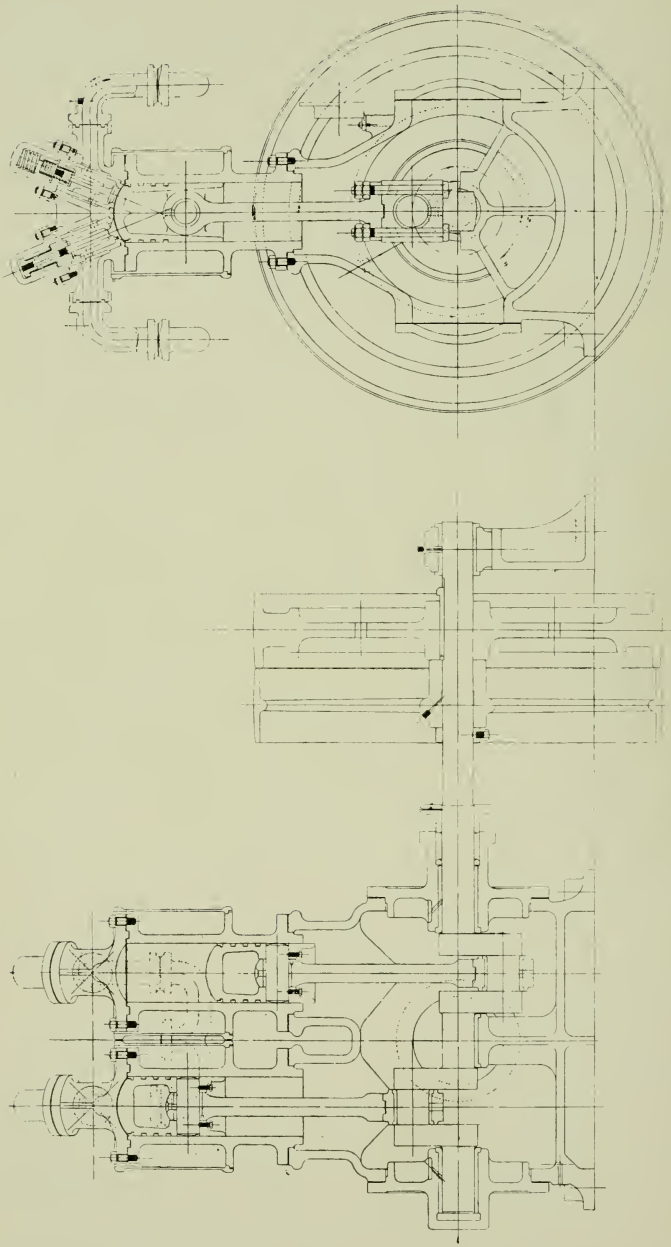

Figure 12

cylinder cover. The object of this design is to ensure safety in the case of excess of liquid ammonia or mechanical obstructions, such as broken valves, entering the cylinder. At such times, when the piston reaches the top of its stroke, the obstruction simply raises the false head without doing any damage.

Figures 14 and 14a show multi-cylinder compressors of this type directly coupled to an electric motor.

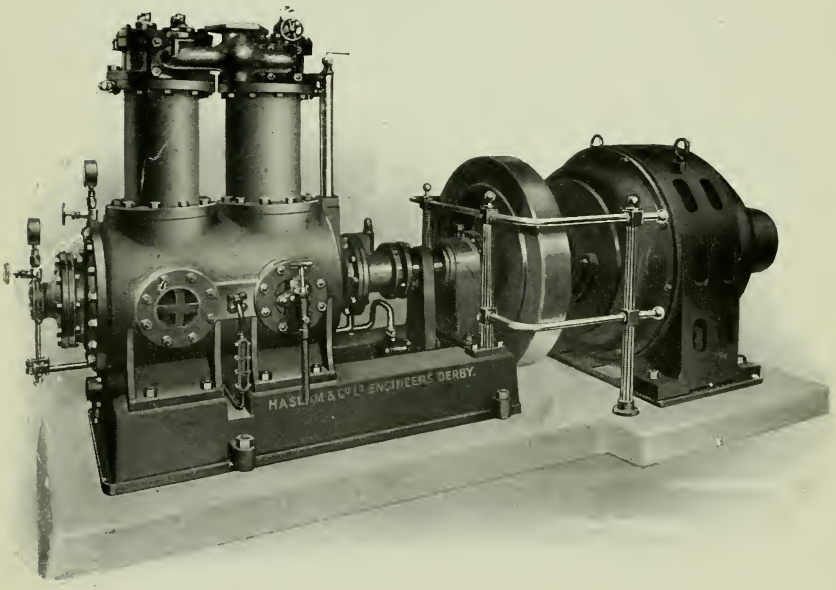

Figure 12a

Both types of enclosed compressor just illustrated suffer from the defect that the ammonia comes into contact with the lubricating oil in the crank-case. Ammonia dissolves in the oil, and whenever there occurs a reduction in the pressure in the crank-case, some of this ammonia is evolved with the production of large quantities of a froth, consisting of bubbles of ammonia, surrounded by films of oil. The production of this froth is aggravated by the rapidly moving parts in the crank-case. In spite of all the precau-

tions taken to prevent it, the author's general experience is that a large amount of this froth, sooner or later, finds its way through the machine, thus carrying oil into the condenser and refrigerator. Oil present in excessive quantities will reduce the heat transmission in the coils and interfere with the good working of the plant.

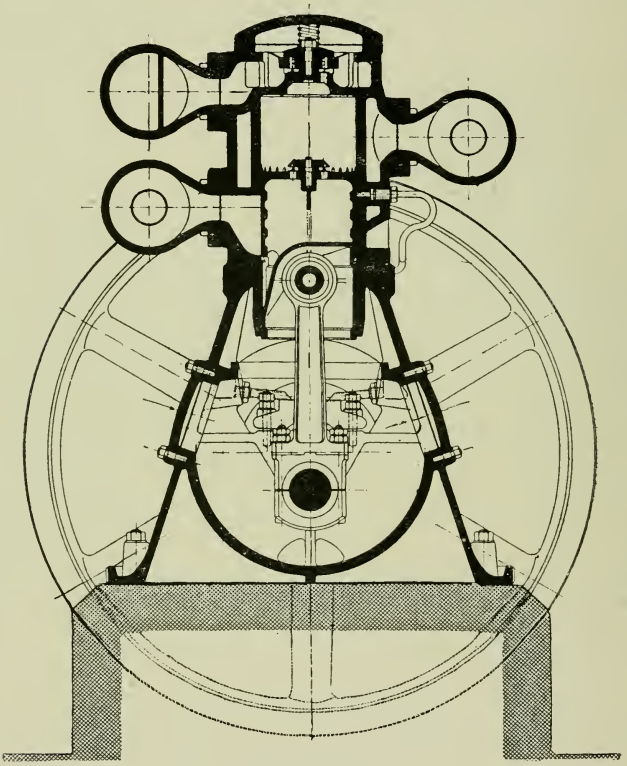

Figure 13

Figure 15 shows a double acting high speed machine of the vertical type. This also has an enclosed crank-case, but the compressor cylinder is entirely separate from and mounted above it. Such machines have a compressor rod working through a stuffing box like the slow speed machines previously mentioned. There is no possibility in this

design of the ammonia coming into contact with the crank-case oil. Double acting machines of this kind cannot be fitted with the false heads which are adopted in the case of single acting machines, and these machines are therefore sometimes said to be dangerous. Experience of some years' running has not resulted in any accident to the author's

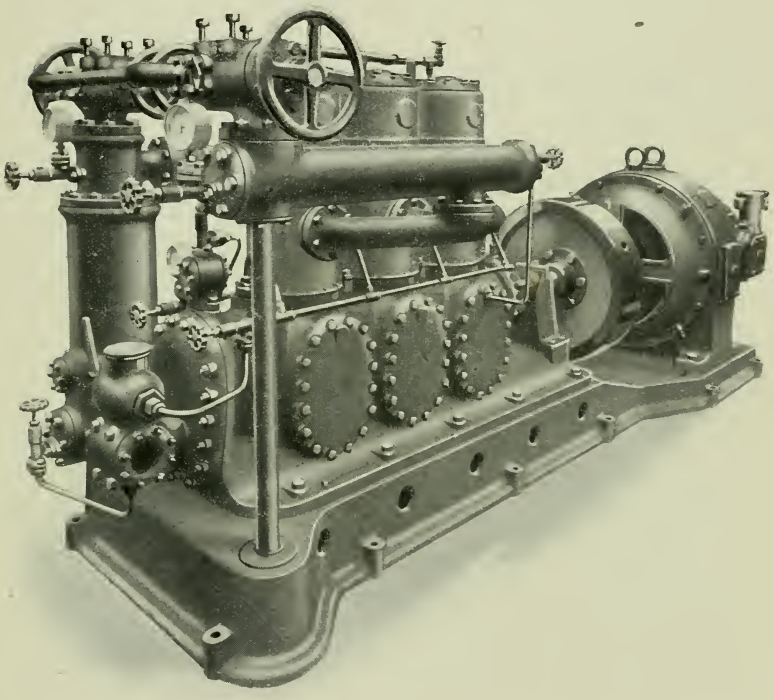

Figure 14

knowledge, and, given a machine of sound design and construction, the danger is no greater than is experienced daily with other types of compressors and in other kinds of machinery.

Double acting high speed machines having forced lubrication and totally enclosed crank-cases are also made in the

horizontal pattern, and give very good results. The floor space taken up is rather more than the vertical type takes, but is much less than required for the slow speed pattern.

Multi-cylinder machines of any type sometimes suffer from the trouble of overheating in some one or more of the cylinders. It is impossible to be sure, where a number of cylinders are drawing from the same suction pipe, that each

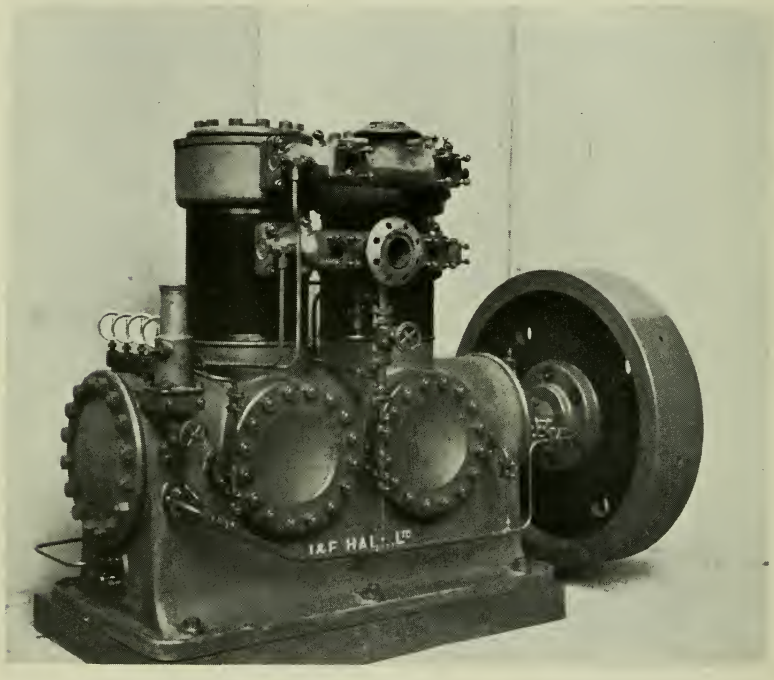

Figure 14a

will get its fair share of the liquid carried in suspension in the vapour. Consequently, any cylinder getting a deficiency of such liquid will work hot, and the overheating may be serious. It is found that the particular cylinder working hot will vary from time to time, sometimes being one cylinder, sometimes another. Double acting vertical machines may also show the same fault unless properly

designed and installed. In such cases we find the top end of the cylinder working hot and the bottom end cool.

High speed CO_2 machines may also be made double acting or single acting. In the double acting type, a compressor cylinder is mounted on a crank-case containing the motion work. The same crank-case and motion work would do equally well for any other type of compressor.

The single acting high speed CO_2 machine illustrated in Figure 16 is of special construction. This machine runs at 400 to 600 r.p.m., and is of the type having the suction valves in the piston, and the delivery valves in a false or

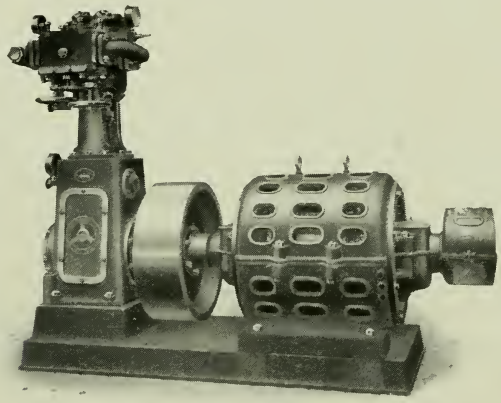

Figure 15

safety head on top of the cylinder. The crank-case is strongly constructed of a steel forging, as it contains CO_2 under considerable pressure. Forced lubrication is adopted, as is now usual in all makes and types of enclosed high speed machines.

In cases where the refrigerator pressure is very low, or the condenser pressure high, that is to say, where the ratio of compression becomes excessive, the compressors should be compounded, and the compression carried out in two stages. With machines having a clearance volume of $2\frac{1}{2}$ per cent., a ratio of compression of 10 can be efficiently

obtained in one stage; beyond this the machine should be compounded. With greater clearance compounding should be adopted earlier.

The outstanding feature of CO_2 for chemical works is its ability to produce easily temperatures lower than are usually met with in ordinary refrigeration work. Thus temperatures down to $-58°$ F. ($-50°$ C.) can be attained

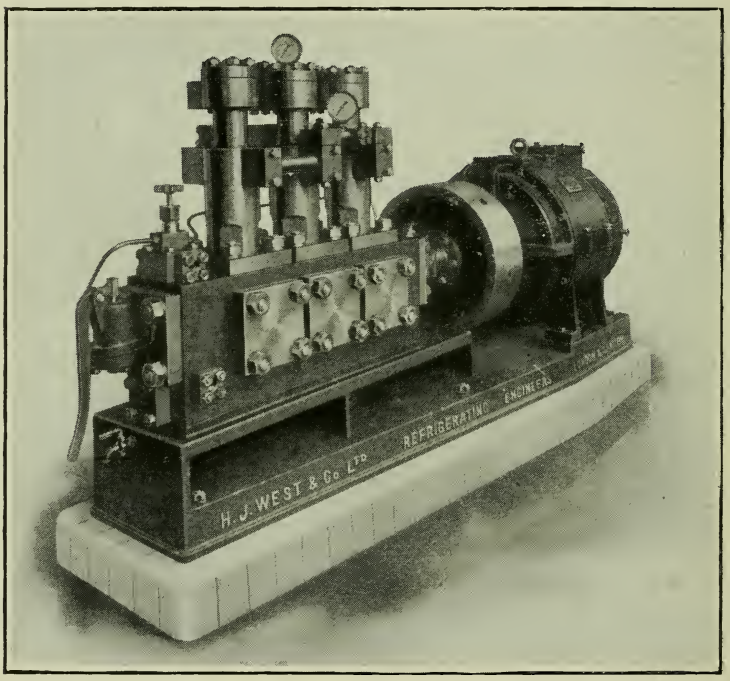

Figure 16

without the pressure of the CO_2 in the refrigerator becoming too low, or the volume of CO_2 vapour to be handled becoming so great as to make the plant unduly large and costly.

At a temperature of $-58°$ F. ($-50°$ C.) the CO_2 would be evaporating under a pressure of 100 lb. per square inch absolute. To compress from this pressure to that in the condenser, a two-stage compressor would, of course, be

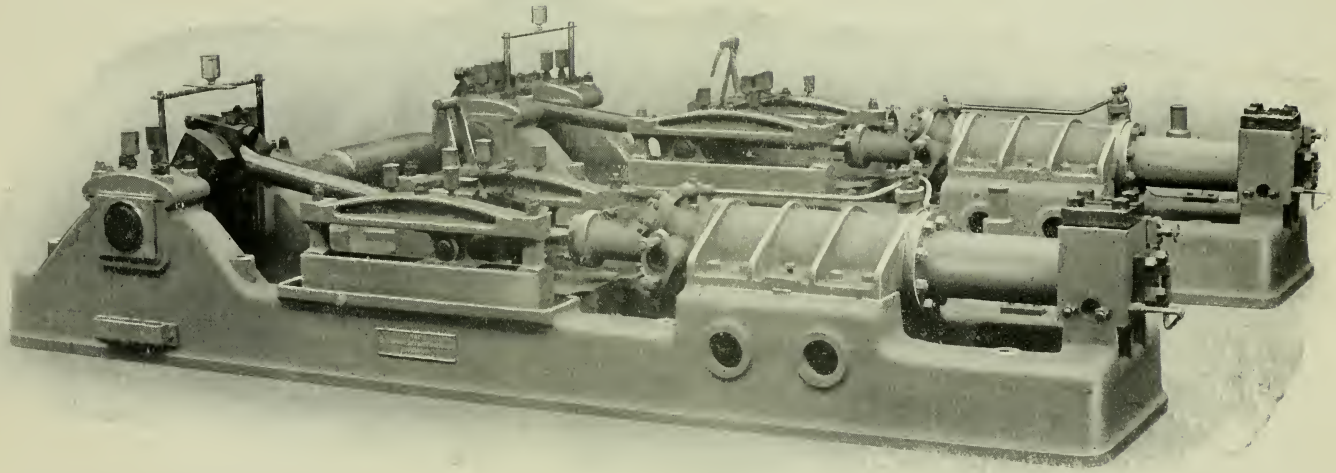

Figure 17

used. Figure 17 shows a large CO_2 machine of this kind, built to the requirements of the author, for working under the conditions just mentioned. This machine, one of four such units, consists of two lines of compressors, each line having one first stage and one second stage cylinder mounted one behind the other. The whole has worked excellently.

Ethyl Chloride Machines.

Machines using ethyl chloride have found little or no application in chemical works. Their use has so far been restricted to very small units. The pressure under which the ethyl chloride must be evaporated is very low. For example, the boiling point of the liquid at atmospheric pressure is 54.5° F. (12.5° C.). To produce any lower temperature than this, the ethyl chloride must be evaporated under a partial vacuum. This fact, together with the large volume occupied by the ethyl chloride vapour rules out the ordinary reciprocating type of compressor. It has only been comparatively recently that a good rotary compressor has been brought out for this class of work, and its manufacture and application have not yet developed it into sizes of service to the chemical industry generally. Its adoption should therefore be undertaken only with expert advice. At the same time it should be noted that a good rotary compressor, where it can be applied, is a more rational design than a reciprocating machine. It has been claimed that ethyl chloride is advantageous for chemical works, in so much that its low temperature permits it to be evaporated directly in, say, acid resisting apparatus of weak mechanical strength, for example, glass tubes and coils. This is true, but the same result can always be obtained with the other refrigerants by first cooling brine and circulating the brine (or other suitable fluid) through such glass tubes, etc. The use of brine involves a double transmission of the cold, and is therefore not so economical. With good design, however, the difference in the power required to operate a given plant either by directly evaporating the refrigerant in such acid-resisting coils, or by using an intermediate fluid such

as brine, would not be generally very great. It must further be remembered that the power required to operate even a large refrigerating machine is not usually considerable, and that the whole cost of power for the refrigerating plant will form only a small proportion of the cost of producing any chemical substance in the manufacture of which refrigeration is used. Too much attention must not therefore be paid to claims such as those stated.

One advantage the ethyl chloride would appear to possess, however. It is good solvent, and is used for oil extraction processes. It is sometimes necessary in such processes to cool the extract in order to bring about crystallization of certain constituents. While usually such cooling must be done by cold brine or water, from a separate refrigerating plant, if ethyl chloride is the solvent the cold may be produced by evaporating off the ethyl chloride under reduced pressure. It is stated that the extracted material may be obtained at as low a temperature as +5° to −4° F. (−15° to −20° C.). It thus provides us with a combined extraction and refrigerating apparatus.

A similar advantage must be accorded to trichlorethylene which, already a recognized solvent, has also been proposed for use as a refrigerant.

Absorption Machines.

We have described in Chapter I the outline of an ammonia absorption plant. Figure 18 shows a plant as actually constructed. It will be seen to consist simply of vessels, and has no moving parts except the small pump for circulating the ammonia liquor. Plant of this type has certain advantages and disadvantages. Firstly, the capacity or the amount of refrigeration produced by absorption machines does not diminish nearly so much as the capacity of a compression machine would do when we have to work at lower temperatures. Also such machines may be operated by exhaust steam from small engines, pumps, etc., which would usually exhaust into the atmosphere. Their disadvantages are that they do not respond to variations in the

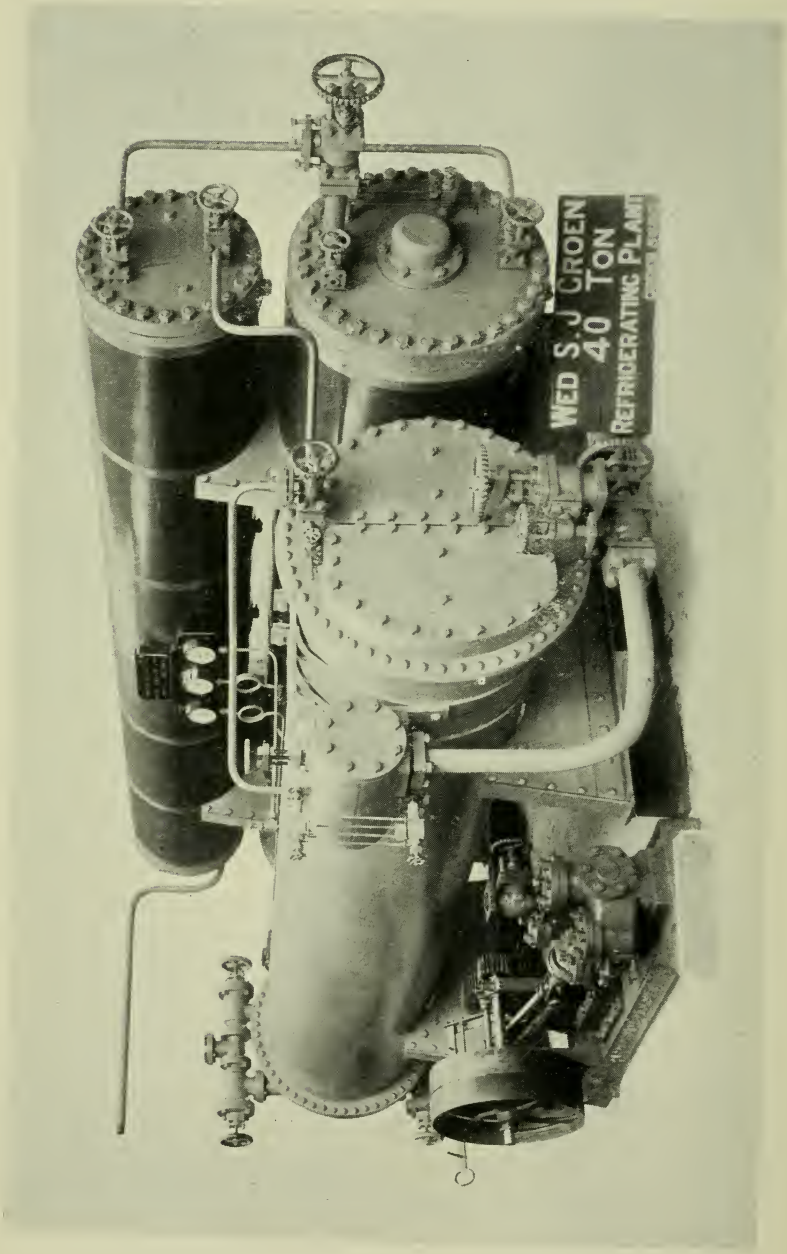

Figure 18

amount of refrigeration required as readily as a compression plant. This is because the absorption plant contains a very large amount of liquor, to which a considerable amount of heat must be added or withdrawn to effect changes in the working conditions. This takes time, hence the comparative sluggishness of absorption plants. Further, the heating of large masses of liquor becomes expensive unless waste steam is available. A much greater supply of cooling water is required for absorption plant than for compression plant. Given, however, a supply of waste steam and a steady load, the absorption plant can be made a very economical proposition. Very few such machines are to be found in this country, however, and recently the best known makers of such plant have given up its manufacture. Its adoption and the obtaining of spare parts and effecting repairs would, therefore, be a matter of some difficulty at present. Absorption machines are more used in U.S.A. than elsewhere.

Water Vapour Machines.

For the production of moderate temperatures, such as are required in the cooling of water, a refrigerating machine using water and water vapour was brought out, but does not seem to have made much progress, at any rate in this country. The cold is produced by evaporating water under a very high vacuum; in fact, the whole success of such plant lies in producing high vacua at reasonable cost. Figure 19 illustrates this type of plant. The refrigerator is simply a vessel in which a high vacuum is maintained by a steam jet ejector *B*. Through the vessel *A* circulates the brine or water to be cooled. Under the high vacuum some of the water evaporates and thereby cools down the remaining fluid. The water vapour removed by the steam jet is forced into a condenser, *D*, in which a rotary air pump *E* creates a vacuum. The vacuum in the condenser depends, of course, on the temperature and quantity of cooling water available, and a large supply of cool water appears to be essential to the success of such plant. The vacuum in the refrigerator is higher than that in the condenser, due

to the action of the steam jet. Thus it is stated that with a pressure in the condenser of 2 inches absolute, the steam jet will give a pressure in the refrigerator of only 0.3 inches absolute. The lowest practical temperature these machines can reach when cooling brine is about 20° F. (6.6° C.).

The following are some figures taken from *Cassiers Magazine* relative to the operation of water vapour machines : —

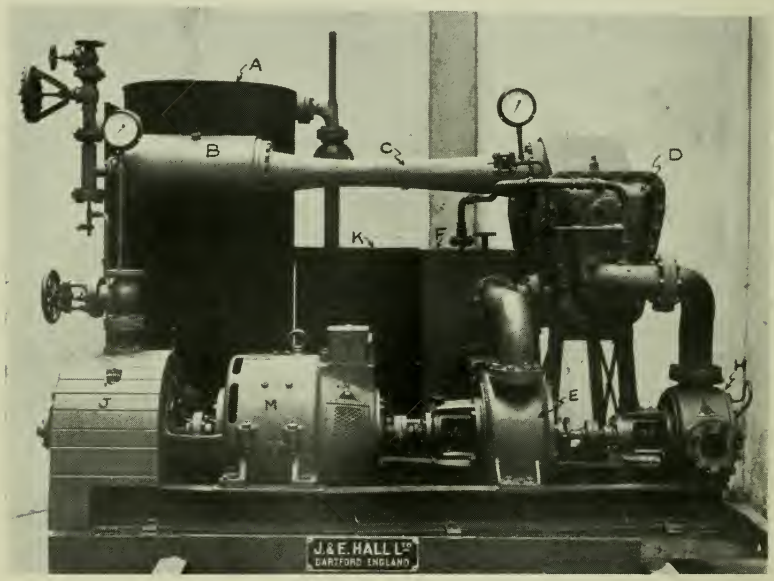

Figure 19

Evaporating temperature 41° F. (5° C.), 360 B.t.u. refrigeration obtainable per lb. of steam used.

Evaporating temperature 27° F. (−2.8° C.), 306 B.t.u. refrigeration obtainable per lb. of steam used.

For each lb. of steam used in the ejector 27.5 lb. of condensing water are required, allowing a rise in temperature of such water from 52° F. (11.1° C.) to 86° F. (30° C.).

A plant of this type was in operation at the Bethune mines, North France, previous to the European War, 1914. This water vapour plant was employed in a benzol recovery process, and cooled brine to a temperature of 41° F. (5° C.). The plant consisted of two units, of a total capacity of 240,000 B.t.u. per hour, with a 15 h.p. motor to drive the various pumps.

The first cost of water vapour plants is stated to be considerably less than vapour compression plants.

Cold Air Machines.

Refrigeration by the cold air machine is now obsolete, because such machines require four to five times the power to drive them that a vapour compression machine would take. In a case of emergency, however, and in a remote country where supplies of ammonia, etc., might not be readily available, it is possible that there may be no alternative but to arrange a cold air machine. In Figure 3, a diagrammatic cold air machine is shown, consisting of a compressor, expansion cylinder, and coolers arranged as one machine. It must be realized that it is not necessary for these various parts to be so combined. In arranging an emergency cold air machine, the compressor may be any ordinary air compressor, or the supply may be taken from the works' compressed air main. The expansion cylinder can be constructed out of an ordinary steam engine with the valves arranged to give an early cut-off, and therefore full expansion of the air. In a proper expansion cylinder the valves would be of the Meyer or Corliss type, of course. The cold air exhausted from the expansion cylinder would then be taken to cool whatever was required. The air supplied to the expansion engine should be as dry as possible, because any moisture present will deposit, during expansion, as ice or snow, and require frequent clearing out. Also the presence of such moisture prevents the air falling to so low a temperature as it would do if dry. Cold air machines usually work with air at a pressure of about 45 lb. per square inch gauge, and expand this

down to atmospheric pressure. Assuming such air, reasonably dry, to enter a good expansion engine at a temperature of, say, 50° F. (10° C.), it would be discharged at a temperature of about −60° F. (−51° C.), after doing work in the cylinder. At this temperature and under normal atmospheric pressure, each cubic foot of air exhausted from the expansion engine would weigh 0.0994 lb., and could therefore do refrigeration equal to 0.236 B.t.u. for each 10° F. (5.5° C.) it was allowed to rise in temperature. The volume of air exhausted from the expansion engine is equal to the volume of the expansion cylinder. Thus, supposing we had arranged an expansion engine with a cylinder 12 inches diameter by 20 inches stroke, running at 150 r.p.m., with air supplied under temperature and pressure conditions previously mentioned, the volume of air discharged per hour would be—

Area of cylinder × stroke × revs. × 2 × 60 = 28,300 cubic feet per hour.
Weight of air discharged 28,300 × 0.0994 = 2,810 lb.

Supposing this air to be allowed to rise in temperature to 20° F. (−6.6° C.), then the refrigeration obtainable would be equal to—

Weight of air × rise in temperature × specific heat
= 2,810 × 20° F. − (−60° F.) × 0.238
= 2,810 × 80 × 0.238 = 53,500 B.t.u. per hour.

This would be the gross quantity of refrigeration, and we should have to make a deduction for heat leakage depending upon the circumstances of the particular case.

Liquid Cooling or Multiple Effect Compression.

Reverting to Chapter 1 we saw that to minimize the loss of refrigerating effect which takes place due to the expansion at the regulating valve, it is necessary to cool the liquid refrigerant as much as possible before it is allowed to pass the valve.

Figure 2 shows diagrammatically an arrangement of plant designed to accomplish this liquid cooling. Such an arrangement has become known as the "multiple effect compression system." Such systems are, of course,

applicable to all refrigerants, but in practice liquid cooling is applied almost solely to CO_2 machines, and not always to these. In the U.S.A. a certain amount of ammonia machinery has been fitted for multiple effect working.

The advantages of the system increase as the condenser and refrigerator temperatures become further apart. Applying such a system to a CO_2 plant, what advantage can we get? In general there results: —

(*a*) an increase in output due to the reduction in the loss of refrigerating effect at the regulating valve;

(*b*) an increase in the coefficient of performance, that is to say, each unit of energy expended produces more refrigeration than it did formerly.

There is, however, considerable difference of opinion as to what are the magnitudes of such increases. It is possible to calculate these increases for any given set of conditions, but the author refrains from giving any such figures here because the improvement shown depends enormously upon the initial conditions chosen to start with. The improvement may thus be made to appear larger or smaller, and the result is very confusing to the non-expert.

Figure 20 shows two sets of test results obtained with multiple effect CO_2 machines. This data is taken from two papers read before the Cold Storage and Ice Association, 1912-13. The one set of results indicated by the full lines claims an increased output for the multiple effect system over the ordinary system of over 130 per cent., with an economy of power per unit of refrigeration of over 50 per cent. The second set of test results, indicated by dotted lines, only show an increase of output of 80 per cent. with some 20 per cent. reduction in the power per unit of refrigeration. Two such diverging sets of results from similar machines under similar conditions of working is not a satisfactory state of affairs. It may be said that both sets of results given are published by interested parties, and no data by an independent authority is yet available. A considerable number of multiple effect CO_2 machines are

now being installed, and it is interesting to note that all manufacturers will now supply such machines if required. There is no doubt as to the advantage of the system as

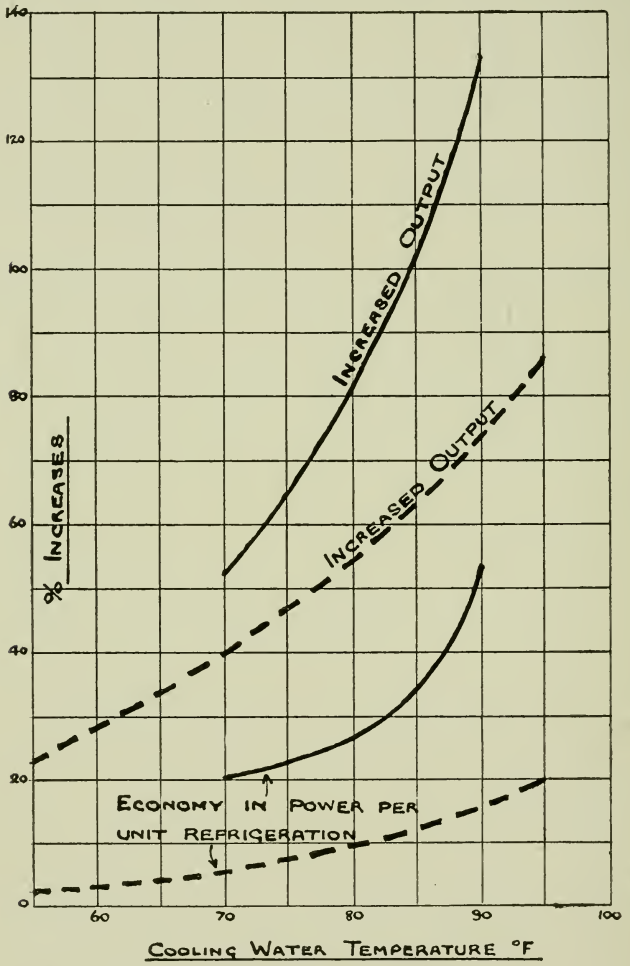

Figure 20

the condensing water becomes warmer, and as the difference between the condenser and refrigerator temperatures increases, but no manufacturer has yet taken full advantage of the possibilities of the system.

It should be noted that the data given in Chapter 3 for calculating the capacity and power required for CO_2 machines relates to those working on the ordinary system indicated in Figure 1, and not to the multiple effect system.

An objection which has been taken to the multiple effect system is that the receiver is liable to become over-full of liquid refrigerant, in which case liquid will be carried into the compressor in excessive quantities, with risk of damage. The right course is to develop a receiver which is reliable and not subject to the above defect. The supply of liquid to the receiver is regulated by hand, while the outlet from the receiver is regulated by an automatic float gear inside the receiver. As a standby a hand controlled outlet is also provided.

Use of an Auxiliary Refrigerating Machine.

In certain instances, which are, however, very infrequent, the necessary cooling of the liquid refrigerant may be performed by an auxiliary refrigerating machine with advantage. Thus, in one case, when the author had to install a very large plant for producing a temperature of $-58°$ F. ($-50°$ C.) there happened to be already on the works a small ammonia machine not in use. The main plant had to be on the CO_2 system, of course, ammonia being entirely useless for such temperatures. The small ammonia machine was, however, pressed into service to cool the liquid CO_2 to a temperature of about 23° F. ($-5°$ C.) before it went to the regulating valve. This expedient increased the output of the main plant over 75 per cent. with an economy of power amounting to 27 per cent. per unit of refrigeration.

Two-Pressure Systems.

In the previous descriptions of multiple effect working, it will have been observed that the liquid CO_2 evaporates at two different temperatures and under two different pressures. The one evaporation takes place in the liquid receiver and the other in the refrigerator itself. Yet one compressor deals with the vapour from both sources. This

has led to the suggestion of a somewhat similar arrangement in such cases where we desire to refrigerate two substances or spaces at two different temperatures, and still only have one compressor. A sketch of such an apparatus is given in Figure 21. In this, the compressor (5) discharges into the condenser coil (6). The liquid refrigerant passes to the main regulating valve (1) and thence into the higher temperature refrigerator (2).

Some evaporation of the liquid refrigerant takes place in this, and the mixture of liquid and vapour passes to a

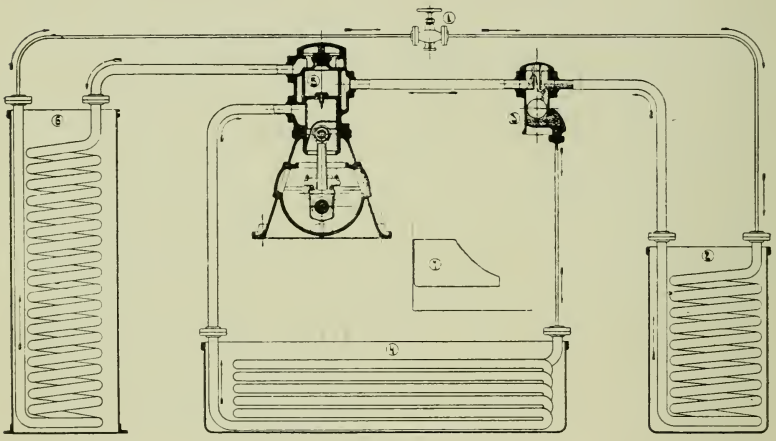

Figure 21

separator vessel (3). From this the vapour goes direct to the compressor, while the liquid passes a float-controlled regulating valve into the lower temperature refrigerator in which evaporation is completed. By such an arrangement the refrigeration is produced with a less first cost than if two compressors were used, while the running cost is less than if one larger machine working at the lower temperature were used. The vapour from the higher temperature refrigerator enters the compressor by the piston uncovering ports in the cylinder wall at the end of its suction stroke. The vapour from the lower temperature refrigerator enters through the suction valves in the usual way.

American Practice.

With regard to American practice, the type of machine apparently now greatly in favour is the vertical single acting type. The machines are of the enclosed pattern with trunk type pistons and safety heads. Some makers, however, do not fit the safety head, and locate both suction and

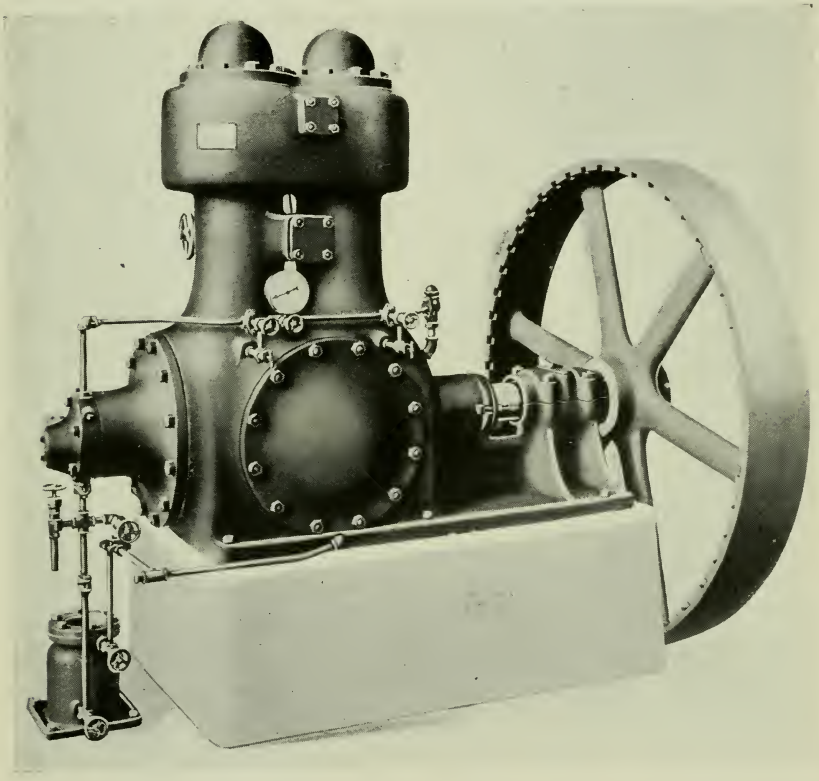

Figure 20a

delivery valves in the cylinder cover, the machines still being kept single acting. An excellent feature of some single acting designs is the over-running of ports in the cylinder wall by the piston at the end of its suction stroke, so that free communication is established between the

cylinder and the suction pipe. This eliminates all throttling of the incoming gas by the suction valves, and avoids the losses indicated in Tables 7 and 8. A distinctive feature of all American machines is the extensive use of

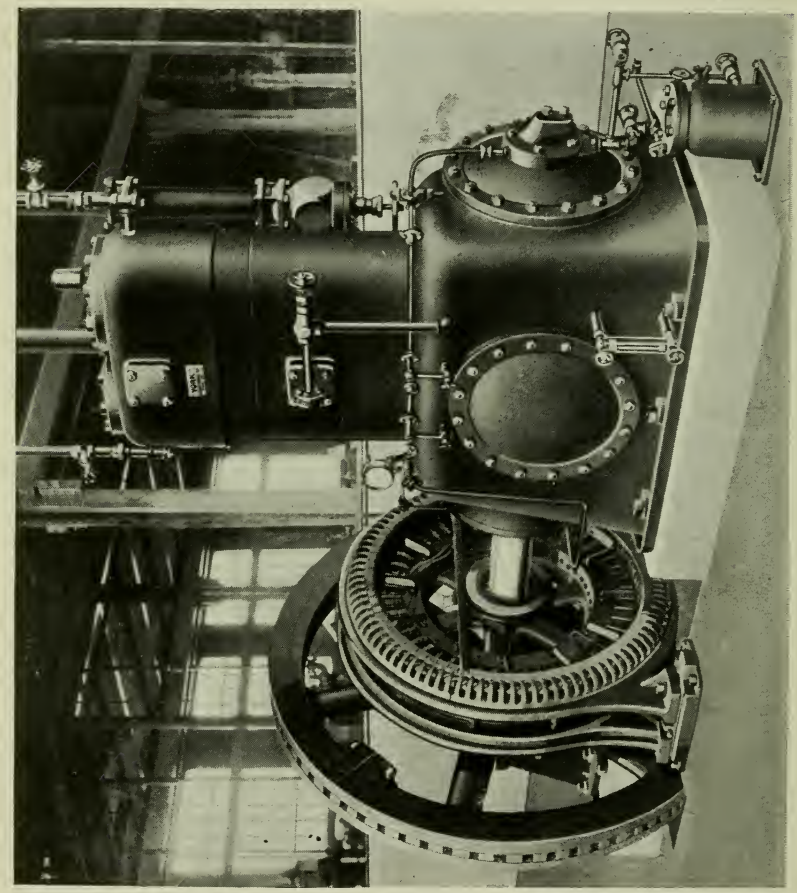

Figure 20b

screwed pipe joints and fittings, compared with the flanged joints used in this country.

Figures 20a and 20b show two American machines by a leading maker.

Wet and Dry Compression.

We indicated, in Chapter 1, two different methods of operation, called "wet" and "dry" compression respectively. The majority of British machines work on a partly wet and partly dry system, while American and Continental builders favour the dry system. The author's experience in operating machines in both ways is, in general, that the most refrigeration is obtained from the plant when the vapour entering the compressor is saturated and carries some liquid in suspension, that is, when working on a partly wet, partly dry, basis. In this connection, it is interesting to note that while American machines were formerly very liberally provided with water jackets for working on the dry system, present day designs show a considerable curtailment in the extent of water jacketing. This is entirely as it should be. In general, more than one half the compression stroke must be accomplished before the compressed vapour is raised to a temperature equal to that of the cooling water supplied to the jacket, and no cooling can be accomplished before such a temperature is attained. Indeed, before this point is reached the action is reversed, the jacket water being cooled by the colder refrigerant in the compressor cylinder. This is a bad feature, as any such heat added to the refrigerant causes it to expand and reduces the output of the machine. No machines should be tolerated which have water jackets round or near to the inlet to the compressor and the whole of the cylinder barrel. Such jackets should be strictly limited to the regions round the discharge valve.

Chapter VI.

CONDENSERS.

Condensers used with refrigerating plant may be of three general types: —

(*a*) Atmospheric or Surface Evaporative.

(*b*) Submerged.

(*c*) Double Pipe.

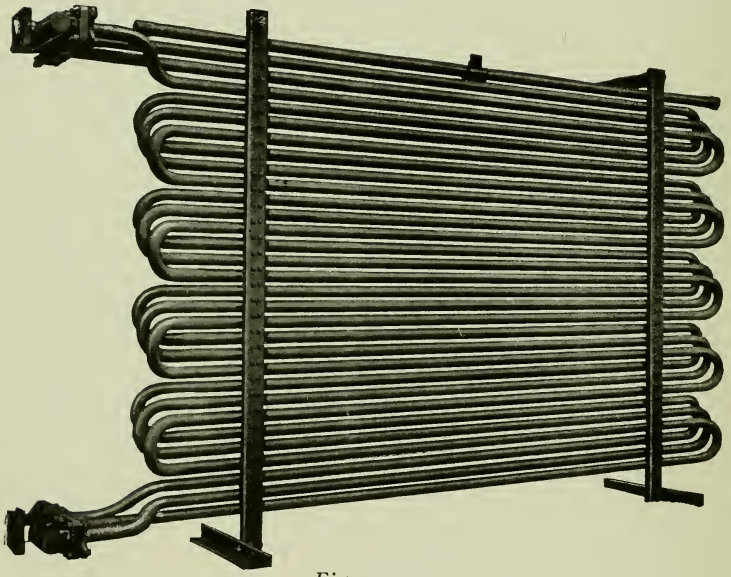

Figure 22

Atmospheric Type.

The first type consists essentially of sheets of pipes (or coils) placed vertically over a tank. Water is sprayed over the outside of the pipes or coils, and collects in the tank, being circulated over and over again by a pump. Such condensers must be placed in the open air, generally on a

roof, and surrounded by a construction of louvre boards. These are so arranged as to keep the direct rays of the sun from striking the coils, and to prevent strong winds blowing the water spray off the coils. At the same time the louvre boards do not prevent a free circulation of air round the pipes. This air circulation causes a small quantity of the water spray to evaporate, thus keeping the remaining water cool enough to permit of its re-use. Figure 22 shows such a condenser (without louvre boards in place), and it will be seen to consist of sheets of pipes, each sheet being made up of interlaced grids. The ends of the pipes are connected into common headers, top and bottom. Each grid is made up of lengths of pipe welded together, and therefore continuous without joints. Formerly such condensers were built up by screwing pipes into return bends or connecting to flanged bends. Such a design involves a multiplicity of joints, and should be avoided. The gas from the compressor is discharged into the top of the condenser, and the liquid taken away at the bottom.

Flooded Condensers.

A variation of the atmospheric condenser known as the "flooded condenser," was introduced some time ago, but does not appear to have made much progress in this country. Such a condenser is illustrated in Figure 23, from which it will be seen that the hot gas is introduced at the bottom of the condenser, while the liquid leaves at the top. The bends connecting the ends of the straight pipes have an internal lip, whose object is to keep each tube partly full of liquid refrigerant. It is claimed that these condensers can be made much smaller for a given duty that the ordinary atmospheric type, and at the same time give increased efficiency. While these claims have been supported by independent authority the inventor's explanation of why such results are obtained is open to question, and such flooded condensers seem to have introduced troubles of their own. Better results can be obtained from the ordinary type by simply introducing the gas at the bottom

and taking away the liquid at the top, due precaution being taken to prevent liquid running back into the compressor.

In all flooded condensers the system must be thoroughly well charged with refrigerant, otherwise it will be impossible to get liquid refrigerant to the refrigerator, and the plant will cease to function.

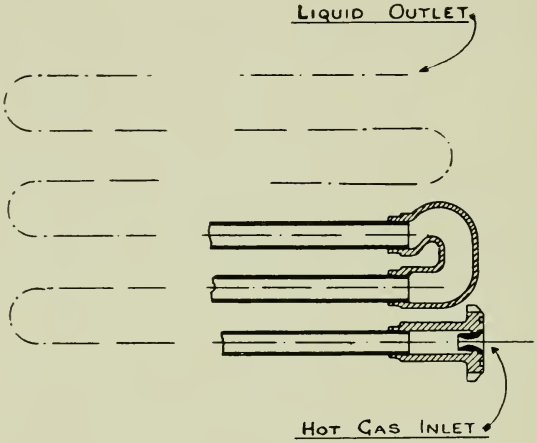

Figure 23

The following figures are a summary of a test of ordinary atmospheric and flooded condensers made by an independent authority. Test A is the atmospheric condenser, while test B is the flooded condenser.

	A	B
Condensing surface	1,786 sq. ft.	596 sq. ft.
Water to condenser	68.5°F	76.1°F
Water from condenser ...	80.0°F	87.7°F
Condenser pressure gauge ...	173 lb.	181 lb.
Corresponding temperature ...	91.8°F	94.1°F
Refrigerator pressure gauge ...	19 lb.	19 lb.
Difference between temperature of ammonia and water ...	(91.8−80) = 11.8°F	(94.1−87.7) = 6.4°F

It will be seen that the flooded condenser has a very much smaller surface than the atmospheric type for the same duty, and therefore occupies less space and is cheaper

in first cost. Also, owing to the smaller difference of temperature between the condensing water and the refrigerant which prevails in the flooded type, the condenser pressure will be lower, and less power will be required to run the plant. This point does not show up in the above test owing to the flooded condenser being served with warmer condensing water than the atmospheric condenser.

Submerged Condenser.

A submerged condenser simply conists of coils placed in a tank through which water is circulated. Such water is used only once, and then goes to waste. Submerged

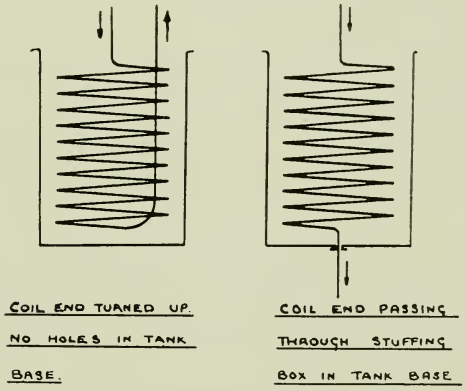

Figure 24

condensers can thus only be used where water is plentiful. They can be placed in any situation and do not require to be placed in the open air. In the case of CO_2 machines the condenser coils can be of copper if the water is corrosive. While more expensive in first cost they reduce expenditure on repairs and renewals, and at the end of their useful life copper coils have a good scrap value. Iron coils may be galvanized for protective purposes.

With coils placed in a tank, the lower coil ends may be turned up and brought to the top of the tank, instead of passing them through stuffing boxes in the tank bottom, as

shown in Figure 24. All joints in submerged coils should be above the level of the liquid, so that they are always accessible.

Double Pipe Condensers.

Double pipe condensers consist of an arrangement of concentric pipes, as shown in Figure 25. The water passes usually through the inner pipe, while the refrigerant is in the annular space between the two pipes. With such a

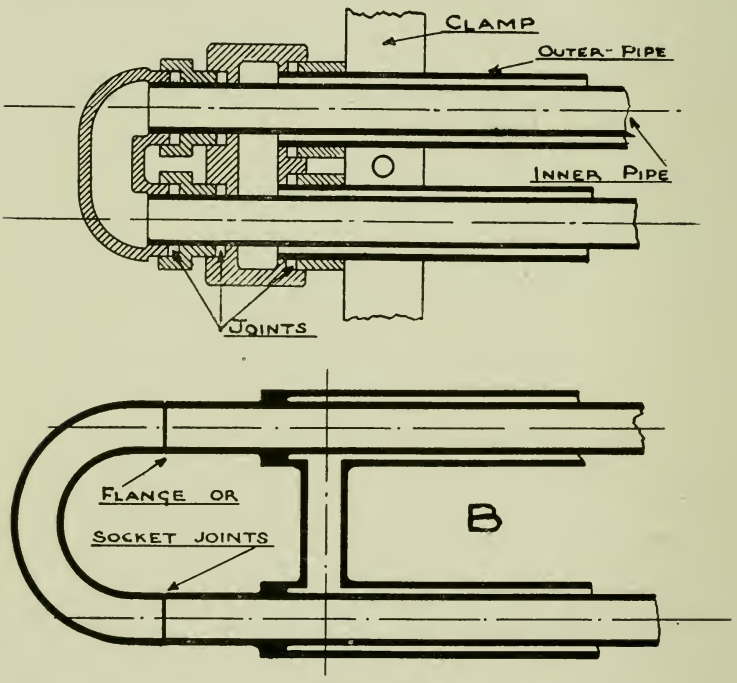

Figure 25

design the refrigerant and the condensing water can be arranged to flow in opposite directions at a greater velocity than usual. Consequently the efficiency is greater, but the apparatus is more complicated, and involves more joints, and is at present only used in small sizes, or in special cases.

One further advantage they possess, namely, the water is totally enclosed in pipes, and may therefore be under

pressure, and so conveyed elsewhere for use after passing through the condenser. Double pipe apparatus is more extensively used in America than in England. Figure 25 (B) indicates another type of double pipe construction in which extensive use is made of welding. This eliminates most of the joints and cheapens the construction, but there is the disadvantage that the annular space is not accessible for cleaning.

Condenser Surfaces.

The amount of surface ordinarily required in condensers may be calculated from the following empirical rules: —

Condenser Surface per Ton Refrigeration (British).

Atmospheric Type.
Surface, 35 to 40 square feet.
Water, 230 gallons per hour.*

Submerged Type.
Surface, 25 to 30 square feet.
Water, 60 to 65 gallons per hour.

Double Pipe.
Surface, 10 to 22½ square feet (mean, 15 square feet).
Water, 72 to 194 gallons per hour (mean, 120 gallons).

Condenser Surface per 10,000 B.t.u. per Hour Refrigeration.

Atmospheric Type.
Surface, 25.7 to 29.9 square feet.
Water, 172 gallons per hour.*

Submerged Type.
Surface, 18.7 to 22.4 square feet.
Water, 45 to 48.5 gallons per hour.

Double Pipe.
Surface, 7.5 to 16.8 square feet (mean 11.2 square feet).
Water, 54 to 145 gallons per hour (mean, 90 gallons).

* This quantity of water is the amount to be circulated over and over again. Additional water for make up purposes of about 2½ per cent. to 5 per cent. of that circulating will be required to make good evaporation losses.

The practice of different manufacturers with regard to double pipe condensers varies very much.

All surfaces given in these rules are calculated on the external diameters of the pipes; and in the case of double pipe apparatus on the external diameter of the inner pipe.

Condensers should be regularly inspected and kept clean. The frequency of cleaning will depend on the water

supply and the quantity of solid matter it carries in suspension. Condensers are so frequently placed in out of the way situations that they are often neglected, and cease to function efficiently.

Condenser Tanks.

These may be made of almost any material, metal, timber, or concrete. Tanks of timber construction may

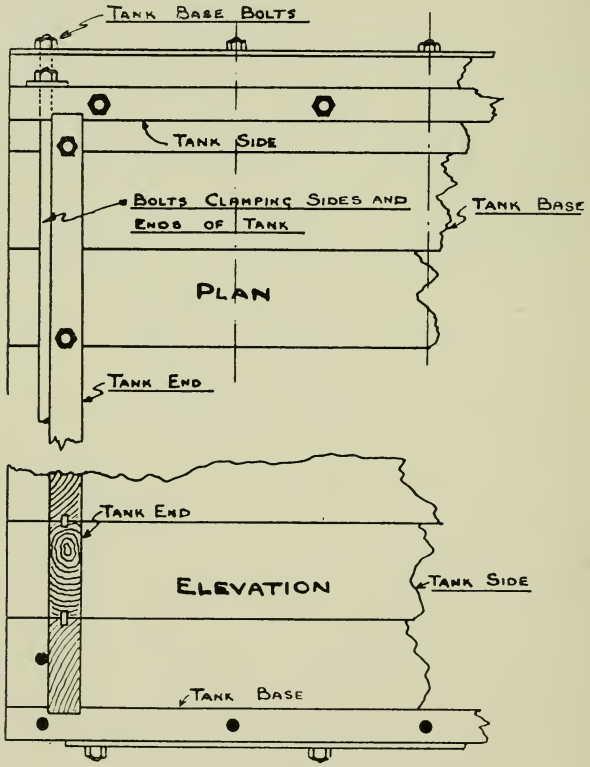

Figure 26

follow the design shown in Figure 26, in which pitch pine planks, grooved and slip tongued, and bolted together, are used. The joints are afterwards finally caulked. Alternatively, a similar construction can be adopted, using

common wood, omitting the tongues, grooves, and caulking, and lining the inside with sheet lead instead.

Condenser Coil Tests.

Condenser coils for ammonia plants are usually tested to 2,000 lb. per square inch by hydraulic pressure to prove their strength; and afterwards tested to 1,000 lb. pressure with compressed air, the coil being submerged in water, if possible. This second test is to detect porosity.

CO_2 condensers would have the hydraulic pressure carried up to 3,000 lb. per square inch.

CHAPTER VII.

COOLING OF LIQUIDS.

In all cases of the application of refrigeration to practical purposes, the first essential is to ascertain what amount of refrigeration is to be done. In the case of liquids the calculation is simple, the net refrigeration to be done being given by—

weight of liquid × fall in temperature × specific heat.

To this quantity we must, of course, add a margin to cover unavoidable losses. The magnitude of such losses will vary enormously with the size of plant and the efficiency with which the apparatus in which the cold is applied, is insulated or lagged.

If the liquid is being cooled with a view to the crystallization or solidification of the whole or any constituent of it, we must also add the heat given up during such solidification, and also any sensible heat removed from the solid. This latter quantity is given, of course, by—

weight of solid matter × specific heat × fall in temperature of solid below its freezing point.

Non-corrosive Liquids.

The simplest case we have to deal with is that of cooling liquids which have no action on the metals used in the construction of the plant. Such liquids are water, calcium chloride brine, etc. In these cases the coolers take the same forms as those previously illustrated for condensers. The most common form is the submerged cooler, which is a replica of the submerged condenser, consisting of coils of pipe placed in a tank. The coils may be either circular in shape or elongated, the latter being generally denoted as oval coils. The cooler may thus be designed to suit the space available to receive it. The tank containing the coils may be open at the top, or closed, depending upon whether the liquid to be cooled is being circulated under pressure

or not. Sometimes in order to increase the efficiency of the heat transmission a set of revolving paddles or agitators is fitted to the cooler so that the liquid is kept in a state of motion over the coil surfaces. If the cooler has an entirely closed tank it should be fitted with a gauge glass and air relief cock, so that the coil surfaces may be kept covered with liquid. Other necessary fittings would be cleaning out doors, drain cock, and relief valve.

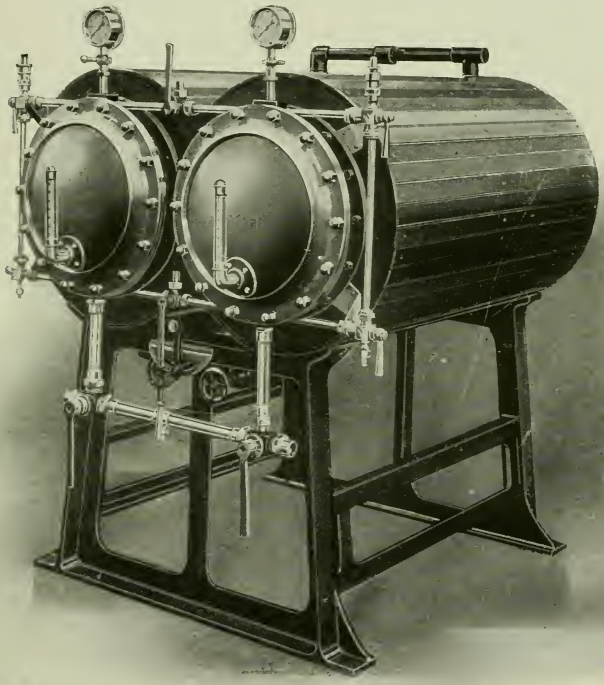

Figure 28

Another very efficient form of cooler for liquids is the double pipe type. It is essential, however, that the liquid to be cooled should be clean and free from suspended matter, which would be otherwise deposited in the tubes.

Coolers similar in construction to the atmospheric condenser have been used, the liquid to be cooled being

sprayed over the outside of the pipes. Such a cooler is often known as a Baudelot cooler. Its chief application has been in brewery work.

Figure 28 shows a cooler specially designed for cooling beer, and consists of two jacketed cylinders fitted with agitating gear. Cold brine is circulated through the jackets. The complete system of thermometers, gauges, and control valves will be noted.

In the case of cooling in order to throw down some one or more constituents in the solid form, as, for example, the cooling of fish oils to throw out the stearine, accessibility

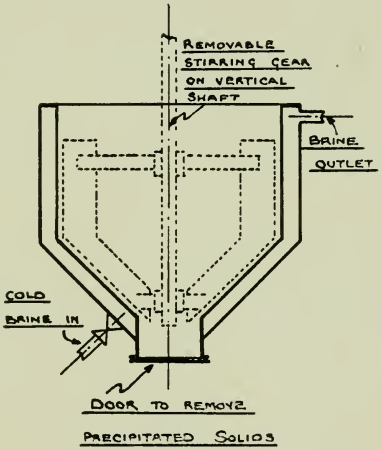

Figure 27

must be the chief feature of the design, so that the solid matter can be readily removed.

Figure 27 shows a jacketed vessel suitable for cooling to effect precipitation. Cold water or brine is circulated through the jacket. A large door is provided at the bottom to remove the solid matter. Stirring gear, consisting of paddles on a vertical shaft, may be fitted, this usually being removable. The liquid remaining on the top of the precipitated matter is removed by syphoning or by a system of decantation pipes.

Another type of cooler rather extensively used is the chill roll. This consists of a hollow revolving roller, the

inside being supplied with cold water or brine introduced through hollow trunnions. If a film of oil is allowed to fall on the roller the oil is rapidly chilled. The precipitated solid matter adheres to the roller, and is removed by a knife or scraper at some point in the revolution. Such apparatus permits of continuous operation. The margarine and paper industries use such plant extensively.

The velocity of the liquid to be cooled over the cold surface should be kept as high as possible. The more viscous the fluid, however, the less the velocity it will be possible to maintain without undue expenditure of power. The more viscous the fluid, therefore, the greater should be the cooling surface provided.

Corrosive Substances.

In the case of corrosive substances, we must arrange for the refrigerating plant to cool water or brine, and circulate this cool brine under little or no pressure through pipes which will withstand the corrosive action of the substance to be cooled. Such pipes may be of lead, or glass, etc. The necessary amount of surface in such cases will generally have to be decided by individual experiment. With submerged coils of lead pipe the heat transmission, and therefore the surface necessary will not be greatly different from that with iron pipes, although the conductivity of lead and iron is very different. The conductivity has little influence in practice, and such factors as the velocity of the fluids, the spacing of the cold pipes, etc., have far more importance. The closer the pipe spacing, the more intimate the contact between the fluid and the cold surfaces. In general, coolers of lead pipe and similar material cannot have such close spacing as with iron and copper pipes, and therefore increased surface should be allowed with them.

Cooler Surfaces.

When cooling water, brine, or similar substances, the following rules provide a check on the amount of surface necessary : —

Submerged Cooler.
Surface, 40 to 65 square feet per ton refrigeration.
or 30 to 49 square feet per 10,000 B.t.u. of refrigeration per hour.

Double Pipe Type.
Surface, 20 square feet per ton refrigeration.
or 15 square feet per 10,000 B.t.u. refrigeration per hour.

Note.—The ton refrigeration here referred to is British ton.

The variation between the maximum and minimum figures is due to different temperature differences between the inside and outside of the pipes. The maximum figure would correspond to a temperature difference of 10° F. (5.5° C.) between the evaporating refrigerant and the cooled liquid leaving the cooler.

With the Baudelot cooler a heat transmission of 150 B.t.u. per square foot per hour per degree F. difference between the inside and outside of the pipes can be obtained.

Ice Making.

A good illustration of the application of refrigeration to produce solidification of a liquid is in ice making. Such a process plays a small but important part in the chemical industry, being adopted in the manufacture of aniline dyes. A few notes on the subject will therefore not be out of place.

Ice manufacture for such purposes as we are now considering is more simple than the manufacture of ice for table use and the cooling of edible products. Many refinements essential for the latter may be omitted when the ice is merely for cooling in a chemical process. It does not matter a great deal, for example, if the ice is clear or opaque; whether the water from which the ice is made is perfectly pure or not, nor whether the ice blocks contain cores. The ice will be broken up and used almost as rapidly as it is produced.

Can Ice.

The most common form of ice making plant at present in use is that known as "can" plant. In this system we have a number of cans which stand, or are suspended in a

tank containing brine. Fresh water, to be frozen, is placed in the cans. Cooling coils, in which the refrigerant evaporates, are located to one (or both) side of the tank, and the brine is kept in circulation over the cans and coils by a propellor or pump. The ice forms on the inside of the cans, and freezing proceeds inwards until the whole of the can contents become solid. In general, it takes considerable time to freeze the last remaining water in the centre of the block, and we frequently find that the freezing operation is suspended before complete solidification results, thus leaving a small hollow space in the centre of the block, called a core. When freezing is complete the cans are lifted out of the brine by a crane, either singly in small plants or in batches in large plants. They are then placed for a few minutes in a "thawing off" tank, containing warm water to loosen the ice from the sides of the cans, and the blocks are then tipped out. If immediate use is not desired the blocks may be stored in an insulated room.

The cans are made of thin galvanized sheet iron, the dimensions of a typical can being 21 inches by $6\frac{3}{4}$ inches at the top, 19 inches by $5\frac{1}{4}$ inches at the bottom, and $49\frac{1}{2}$ inches deep. Such a can would produce a block of ice weighing about $1\frac{1}{2}$ cwt., and of a mean thickness of approximately 6 inches, hence it would be called 6-inch ice.

The thicker the ice the longer it takes to freeze, and under the usual working conditions for this country the following would represent the approximate times of freezing for various thicknesses of ice: —

Thickness of ice, inches...	4	6	8	10	12
Time for freezing, hours	10	21	36	55	78

The freezing time is, of course, obviously dependent upon the temperature of the water initially, and on the temperature of the brine. A low temperature brine gives quicker freezing, but is less economical. A large number of cans for a given output increases the economy by allowing a longer freezing time, and thus permitting a higher temperature brine to be used. Thus, if we have ten 300 lb. cans per ton of ice required, a brine temperature of about

7° F. (−13.9° C.) would be necessary, while with eighteen 300 lb. cans per ton brine at 18° F. (−7.7° C.) would suffice. Of course, the larger number of cans means a large tank, and a greater first cost. A balance must be struck to suit the individual case.

The amount of refrigeration to be done in ice making is about 240 B.t.u. per lb. of ice made in England, increasing to 290 B.t.u. or more in hotter climates. This figure covers the cooling of the water, the freezing of the water, the subsequent cooling of the ice, and allows for such heat leakages as would be experienced in well-designed plants of moderate size. Small plants or badly designed ones would have greater heat leakage and require proportionately more refrigeration.

Plate Ice.

A system producing very good ice in large pieces without the use of cans is that known as the "plate ice" system. In this the water to be frozen is in contact with large cold flat surfaces, and is thus frozen into large flat pieces called plates. A typical size of plate would be 12 feet wide by 9 feet deep by 12 inches thickness, such a block weighing over three tons. It must be cut up for use, and takes a long time to freeze, possibly some 300 hours, and therefore we find plate systems becoming obsolete.

A modification of such a system has been brought out and called the Pluperfect system. In this the flat cold surfaces are dispensed with, and rows of special vertical U-tubes arranged across the tank. The refrigerant evaporates inside these tubes, freezing taking place in all directions round them until the ice produced by adjacent tubes meets, thus joining up and freezing into a continuous slab. A common size of such a slab is 6 feet by 3 feet 9 inches, and would weigh about 11 cwt. When thawed off, the slab has a series of holes running through it, but this is immaterial for our purpose. The thawing off is done by turning comparatively warm refrigerant from the compressor and condenser into the U-tubes.

The Pluperfect Co. state that this system will produce on the average 17 to 20 tons of ice per ton of coal used, and that this has been exceeded by a plant in actual operation, giving 28 tons of ice per ton of coal. An electrically driven plant has given one ton of ice per 56 units of electricity.

A small plant of the author's on the ordinary system, worked by CO_2 compressors and suction gas engines gave 17 to 19 tons of ice per ton of anthracite over the whole year's working.

Usually about 125 to 150 feet of $1\frac{1}{4}$ inch piping are used in the ice tank for each half ton of ice to be made. The ice making capacity is expressed in tons per 24 hours.

Chapter VIII.

THE COOLING OF GASES AND VAPOURS.

As in the case of liquids, the first essential is the calculation of the amount of refrigeration to be done. As a gas is simply a vapour in a highly superheated condition, the calculation resolves itself into

(1) calculation of the superheat (or sensible heat) to be removed from the gas, which is equal to

weight of gas × fall in temperature × specific heat.

The specific heat of the gas will, in general, be that under constant pressure as the cooling will take place under at least an approximately constant pressure.

If we remove all the superheat, the gas becomes a dry saturated vapour, and further cooling will cause condensation to take place.

(2) we should thus remove a further quantity of heat equal to

weight of gas × latent heat of evaporation.

When all is condensed if we cool the liquid further we should remove some sensible heat from the liquid calculated as in Chapter 7.

Air Cooling.

Probably the gas which has to be most frequently cooled in refrigeration work is air from which we remove a portion of its superheat. This forms an interesting typical case because we have to cool air containing more or less water vapour, some of which condenses. In fact cooling of air by refrigeration is the readiest and probably the cheapest method of drying the air when large quantities have to be dealt with. Drying by cooling can, of course, be carried out on any gas quite as well as air.

In the cooling of air it is not usually realized that often more than half the total cooling to be done is required to

condense the moisture, as the following numerical example will show.

We will take the case of having to deliver 250,000 cubic feet of air per hour at a temperature of 35.6° F. (2.0° C.), the initial temperature being 86° F. (30° C.), and the relative humidity of the air being initially 80 per cent. That is to say, the air contains 80 per cent. of the amount of moisture necessary to saturate it at 86° F.

Firstly, it should be noted that if we have to deliver 250,000 cubic feet of air at 35.6° F. (2.0° C.), we must take a larger volume into the plant. The volume is increased in the ratio of the absolute temperatures at entrance and exit, and there is a further increase due to the change in the pressure exerted by the water vapour at the initial and final temperatures. This difference in vapour tension is equal to the maximum vapour tension at the initial temperature multiplied by the initial relative humidity, less a similar calculation for the exit conditions. The air will be saturated at exit, and in our example the change in vapour pressure will therefore be

$$(0.0415 \times 0.8) - 0.007 = 0.0262 \text{ atmosphere.}$$

Therefore the volume entering the cooler must be

$$\frac{250{,}000 \times 303 \times \quad 1}{275 \quad (1 - 0.0262)} = 280{,}000 \text{ cubic feet.}$$

We have assumed that the air is at atmospheric pressure at exit from the cooler.

At a temperature of 86° F. (30° C.) and 80 per cent. relative humidity we find from hygrometric tables that each cubic foot of air entering the cooler would contain 0.001505 lb. of water vapour. As the cooling proceeds, the temperature falls until a point is reached at which the quantity of water vapour present is sufficient to saturate the air. Beyond this point the water vapour condenses out as the temperature falls, the air always now remaining saturated. When the final temperature of 35.6° F. (2.0° C.) of our example is reached, each cubic foot of air leaving the plant would contain 0.0003482 lb. of water vapour.

The weight of water entering the cooler is therefore $280{,}000 \times 0.001505 = 422$ lb. per hour, and the weight of water leaving the cooler is $250{,}000 \times 0.0003482 = 87$ lb. We thus remove from the air $422 - 87 = 335$ lb. of water per hour.

Assuming the average latent heat of evaporation of the moisture to be 1,100 B.t.u. per lb., then the heat removed during condensation is approximately

$$1{,}100 \times 335 = 369{,}000 \text{ B.t.u. per hour.}$$

To this must be added the sensible heat removed from the vapour before condensation and the sensible heat removed from the liquid after condensation. An approximation only to these quantities can be made, because condensation takes place at a range of successive temperatures. We may assume that the whole of the vapour is cooled to a temperature midway between the final temperature and the dew point of the entering air, then a portion condenses, and that the condensate is cooled from the mid temperature to the final temperature. That portion of the vapour which does not condense is, of course, cooled over the entire range of temperature.

In our example the dew point of the entering air would be nearly 81° F. (27.2° C.). Therefore the midway temperature would be about 59° F. (15° C.).

The cooling of the vapour would therefore take

$$422 \times (86 - 59) \times 0.48 = 5{,}480 \text{ B.t.u.}$$

and the cooling of the liquid would take

$$335 \times (59 - 35.6) = 8{,}170 \text{ B.t.u.}$$

The uncondensed portion of the vapour weighs 87 lb., and we have already reckoned this to be cooled to 59° F. (15° C.). Therefore to cool it through the remaining range of temperature, that is, down to 35.6° F. (2.0° C.) would take

$$87 \times (59 - 35.6) \times 0.48 = 978 \text{ B.t.u.}$$

The total cooling done on the moisture is thus 383,600 B.t.u. per hour.

Now the weight of air entering the cooler can be found from the formula

$$PV = RT.$$

where P = pressure in lb. per square foot exerted by the air
= barometric pressure − tension of the water vapour.

V = volume of 1 lb. of air in cubic feet.

T = absolute temperature of entering air
= temperature F. + 461.
= temperature C. + 273.

R = constant = 53.2 with Fahrenheit temperatures.
= 95.8 with Centigrade temperatures.

$\frac{1}{V}$ = weight of 1 cubic foot of air.

In the given example

$$\frac{1}{V} = \frac{14.7(1-0.0332) \times 144}{53.2 \times 546} = 0.0703 \text{ lb.}$$

Therefore the weight of air entering the cooler is $280{,}000 \times 0.0703 = 19{,}700$ lb. per hour.

The heat removed in cooling

weight of air × temperature fall × specific heat.
$19{,}700 \times (86 - 35.6) \times 0.238 = 236{,}000$ B.t.u. per hour.

The total net refrigeration to be done would therefore be $236{,}000 + 383{,}600 = 619{,}600$ B.t.u. per hour, of which 38.1 per cent. is absorbed in cooling the air itself, and 61.9 per cent. in condensing the moisture out of it.

A more correct estimate of the cooling necessary to remove the water vapour can be made by reference to a chart giving a graphical representation of the properties of the air and water vapour. A good example of such a chart is that evolved by Mr. W. H. Carrier, and called by him a "Psychrometric Chart." In the absence of a chart, however, the calculation can be made as shown, and will be generally within 3 per cent. of the result given by the chart. This is a quite sufficient degree of accuracy for practical calculations.

If we had been dealing with any other gas than air the same calculation with regard to the moisture would hold. This is because the amount of water vapour which can

exist in any space depends upon the temperature only, and is the same whether the space be empty (vacuous) or already be occupied by any gas.

Air Drying.

The problem presented to us might have been in a different form. Instead of requiring a supply of cold air we might have done the cooling merely to dry the air, and our example would become as follows: —

We require a supply of 250,000 cubic feet of air per hour at, say, 75.2° F. (24° C.) containing not more than 0.00015 lb. of moisture per cubic foot. The relative humidity of such air would be about 11 per cent., and it would therefore be very dry. We will assume the initial temperature of the air 86° F. (30° C.), and the relative humidity 80 per cent. as before.

From the hygrometric tables we see that to effect this drying we must cool the air to about 17.6° F. (−8° C.), at which temperature each cubic foot would hold 0.0001672 lb. of moisture when saturated, as it would be leaving the cooler.

The volume of air entering the cooler would be, allowing for changes in temperature and vapour pressure

$$\frac{250{,}000 \times 547}{536} \times \left(\frac{1-(0.11 \times 0.0292)}{1-(0.8 \times 0.0415)} \right) = 263{,}000 \text{ cubic feet per hour.}$$

If such air is allowed to warm up out of contact with any source of moisture, the weight of moisture present will not alter, but the air would expand, and each cubic foot at 17.6° F. (−8° C.) would become 1.12 cubic feet at 75.2° F. (24° C.). We have here neglected the change in pressure of the water vapour, but as the air is practically dry, this is negligible. The moisture content of the air leaving the plant, when warmed up to the required temperature, would be 0.0001672 ÷ 1.12 = 0.000149 lb. per cubic foot, or just within the given limit.

The calculation for the amount of cooling to be done may be carried out as in the first case, but there is this

important difference in operation. We are now going to cool the air below the freezing temperature of the water. Therefore while at the beginning of the operation moisture will be removed in the liquid form as water, towards the end of the cooling we shall be freezing the moisture and depositing it as snow or ice. This will accumulate on the cooling coils, and will ultimately necessitate stopping the plant for its removal. In such cases we therefore arrange the cooling to be done in two stages; first from the initial temperature down to, say, 35° F. (1.7° C.), and then in another cooler from 35° F. (1.7° C.) down to the final temperature required. The bulk of the moisture (over 70 per cent. in the given example) is deposited in the first cooler in liquid form, and only the small remaining quantity of water removed in the second cooler appears as ice.

The formation of ice in the second cooler may be obviated by spraying brine on the outside of the pipes. The brine must be of such a density that it will not freeze at the lowest temperature met with in the plant. The brine is circulated over and over again, but by reason of the water from the air which continually mixes with it, it becomes weaker and must be concentrated from time to time. The concentration is effected by boiling off the surplus water or by running some of the weak solution to waste and adding fresh calcium chloride.

In such a case as we are now considering, the cold dry air leaving the plant would be utilized to give a preliminary cooling to the air entering the plant, thus effecting economy in operation. In fact, under suitable conditions, the regenerative action of the cold air is capable of accomplishing some 25 per cent. of the total cooling to be done.

It is also economical and useful to arrange plants of this character on the two-pressure system described in Chapter 5.

Air Coolers.

The usual construction adopted for air coolers is to have a series of vertical sheets of pipes, similar to those used in the atmospheric condenser, but with the sheets closer

together. Such a battery of pipes is placed over a tank, which collects any moisture condensed out of the air, and also holds the brine, which is sprayed over the pipes. Where no brine spray is required the tank may be dispensed with, and the floor of the chamber in which the cooler is housed made watertight, as by asphalting, or covering with sheet lead. In the case of air coolers the chamber containing the coils may be constructed of metal, timber, brickwork, etc. The air is circulated over the coils by a fan situated at one end of the cooler. The refrigerant evaporates inside the coils.

Alternatively, the coolers may be arranged having the coils of pipe in which the refrigerant evaporates in a tank, and surrounded by brine. Over the tank vertical sheets of pipes are fitted through which the cold brine from the tank is circulated. A small quantity of brine is also sprayed down the outside of the vertical sheets, and the air is blown over the sheets as before. Such a design involves a double transmission of heat, first from the air to the brine, and then from the brine to the refrigerant. The refrigerant must therefore evaporate at a lower temperature than if a single transmission of heat only occurred, and the plant is more costly and not so economical. The design has, however, the advantage that there is a large reserve of cold stored up in the brine in the system, and cooling of the air may be carried on for a considerable time after the refrigerating machine has been stopped. This permits minor repairs, adjustments, cleaning, etc., to be done without seriously interrupting the cooling. The brine, in such a case, also smooths out sudden fluctuations in the load on the machine. With coolers in which the refrigerant evaporates directly inside the pipes over which the air is circulated, if anything happens to stop the air circulation, such as stoppage of the fan, the refrigerant in the pipes does not receive sufficient heat to evaporate it. It consequently accumulates in the pipes in the liquid form until a point is reached when large quantities of the liquid enter the compressor and may cause serious damage.

Figure 29 shows a typical construction for an air cooler.

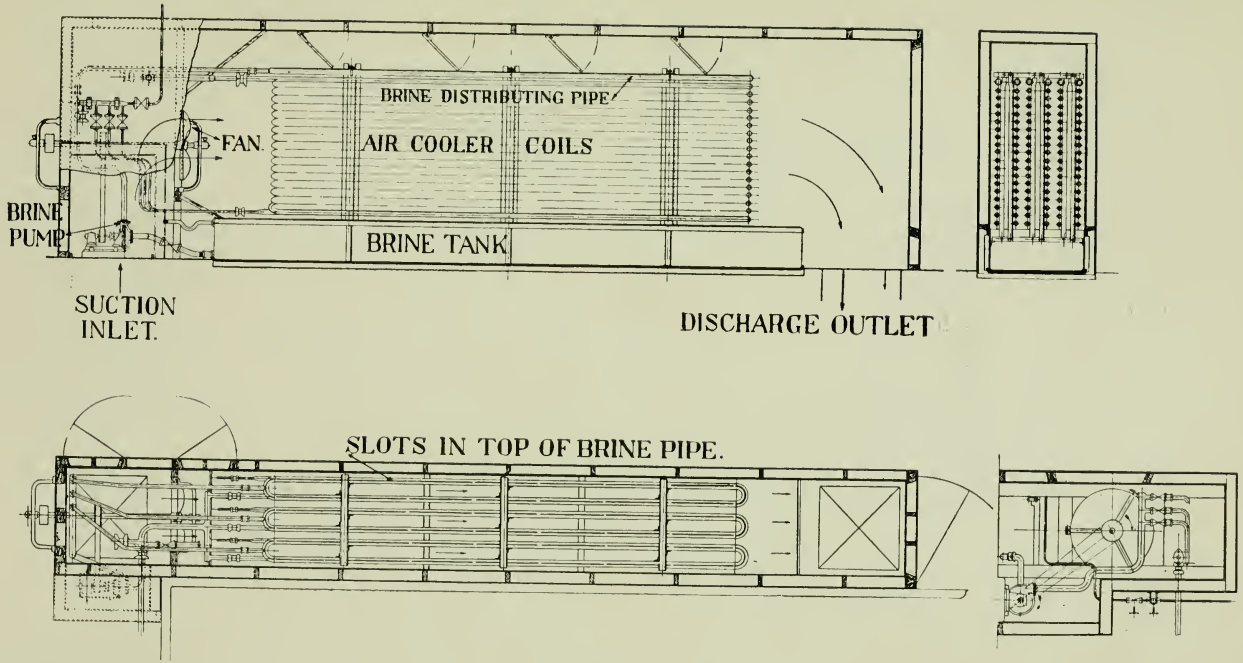
BRINE DISTRIBUTING PIPE
FAN.
AIR COOLER COILS
BRINE PUMP
BRINE TANK
SUCTION INLET.
DISCHARGE OUTLET
SLOTS IN TOP OF BRINE PIPE.

Figure 29

Spray Coolers.

We may also cool air by passing it through a chamber in which nozzles maintain a fine spray of cold water or brine, instead of passing it over cold pipe surfaces. The spray method is much the more efficient, providing the spray is fine enough, as it gives a more intimate contact between

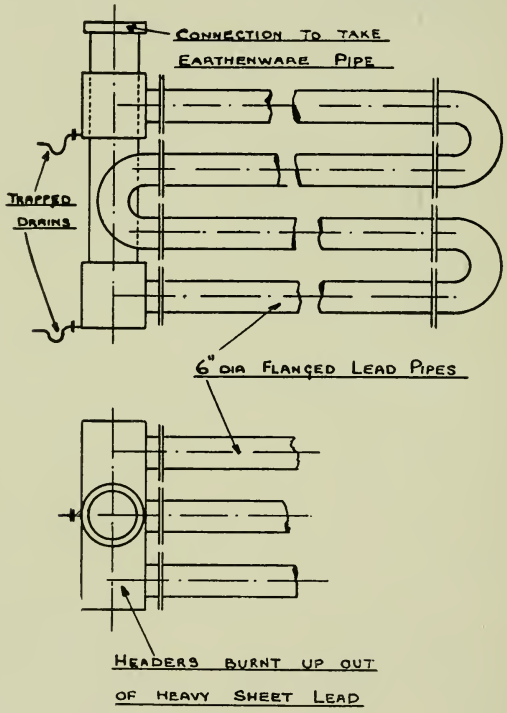

Figure 30

the air to be cooled and the cooling medium than any surface cooler can effect. To prevent the air carrying away with it particles of brine or water in suspension, it is passed through a chamber containing baffles. Both the spray and the baffle chambers can conveniently be constructed over the top of the tank, which contains the refrigerator pipes cooling the brine or water used for the spray. The tem-

perature of the outgoing air is automatically controlled and kept constant at any predetermined figure. The whole arrangement is very compact and complete, and occupies less space than a surface cooler.

Corrosive Gases.

In the case of a corrosive gas, as, for example, wet chlorine, iron cooling pipes are an impossibility. Gases of this nature can, however, generally be dealt with in apparatus constructed of lead. An effective cooler can be constructed by circulating the gases in large diameter lead pipes submerged in a brine tank, or having cold brine sprayed over them. The lead pipes are flanged and connected to headers built up out of heavy sheet lead, suitable supports being arranged to take the weight of the upper parts of the apparatus. The whole cooler can then be readily dismantled for internal cleaning and repairs. Such a cooler is indicated in Figure 30.

Condensation of Gases.

In some instances we require to cool gases in order to effect their liquefaction. Such cases can be divided into two classes: —

(1) those in which the gas to be condensed is pure, that is, contains no appreciable quantity of any other gas;
(2) those in which other gases are present.

The first case presents no difficulty. If the gas is under considerable pressure we may pass it into coils of pipe submerged in cold water or brine. The design of apparatus simply follows that of the condensers used in refrigerating plant.

If the gas is under a light pressure it may be condensed in a metal casing containing cold pipes. In the case of gases which are noxious or corrosive if they escape into the atmosphere, it is advisable to effect their condensation under atmospheric pressure whenever possible. The design in such cases may take the form of that for light pressures, or

in large sizes a chamber of sheet lead, supported on a timber framework, may be constructed and cold pipes fitted inside. The temperatures worked to will, of course, depend upon the physical properties of the substance to be cooled, and the materials of which the cooler is constructed must not be affected by the gas. In almost all cases of liquefaction the gas is previously dried, and frequently a gas which is corrosive under ordinary conditions where moisture is present, is not corrosive when thoroughly dried. The condensation of gases is therefore generally a much simpler problem than that of cooling a moist gas.

The case of an impure gas is more difficult. The presence of the impurity affects the temperature and pressure conditions under which condensation can be effected. For example, pure chlorine will condense at $-31°$ F. ($-35°$ C.) under atmospheric pressure. But a mixture of half chlorine, half non-condensable gas (hydrogen or air) as is sometimes obtained from electrolytic cells, would need to be cooled to about $-58°$ F. ($-50°$ C.) before liquefaction of the chlorine would take place. This is due to the fact that while the mixture of chlorine and non-condensable gas is at atmospheric pressure, the chlorine only contributes a part of such pressure, and we must reduce the mixture to a temperature depending upon the partial pressure exercised by the chlorine.

In cases where there is a non-condensable constituent present means must be provided for removing it. We cannot adopt the simple closed chamber design, for then the non-condensable constituent would accumulate in the chamber to such an extent as to put the plant out of action. The non-condensable gases must therefore be continually withdrawn.

Also it will not be possible to condense all the condensable constituent. Some of it will pass away with the non-condensable part, the composition of the mixture leaving the cooler depending upon the temperature to which the mixture is reduced. Thus, again considering our example of a chlorine mixture of composition, say, 87 per cent. chlorine and 13 per cent. other gases entering a cooler

at atmospheric pressure and temperature, and cooled down to −58° F. (−50° C.), the composition of the gases leaving the cooler would be of the order of 53 per cent. chlorine and 47 per cent. non-condensable gas. It should be noted, however, that this does not mean that we are only condensing a small portion of the chlorine. The gas analyses at

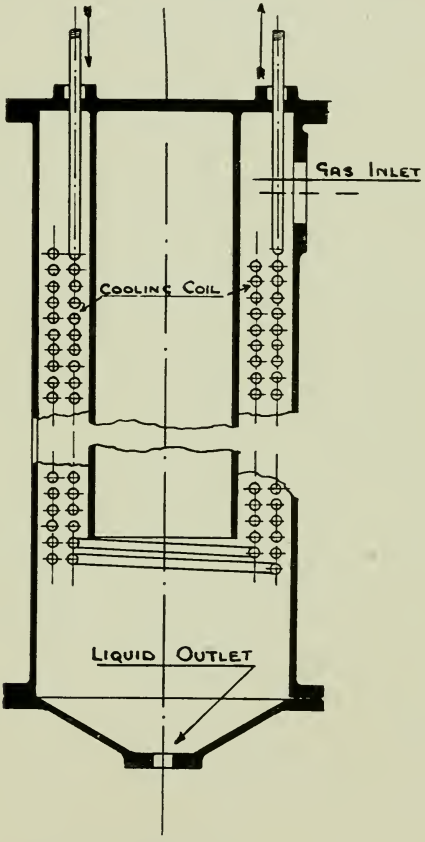

Figure 31

inlet and outlet enable us to determine what this amount is relatively. Thus, with the compositions given, we have in every 100 volumes of mixture entering the cooler, 87 volumes of chlorine and 13 volumes of non-condensable

gases. These 13 volumes pass through the cooler unchanged, and constitute 47 per cent. of the mixture leaving the cooler. The other 53 per cent. must therefore be represented by 14.7 volumes of chlorine, and consequently we have condensed 87—14.7, or 72.3 volumes of chlorine out of every 87 entering the cooler. That is, 83 per cent. is condensed. These points, of course, apply to mixtures of other substances under similar conditions.

Types of Gas Condensers.

Condensers in which the gas is under considerable pressure require no illustration, as they can follow the designs of the condensers shown in Chapter 6.

Figure 31 illustrates diagrammatically a condenser of small size for dealing with a gas under light pressure. This has a metal casing containing two cooling coils. The interior space where there are no coils is blanked off by a metal cylinder so that the gas to be condensed must come into intimate contact with the cooling surfaces. The two cooling coils are arranged in series, but in larger sizes where more coils are employed the bottom coils ends may be brought out through stuffing boxes in the bottom cover, or alternatively turned up into the central empty space in the cooler and brought out of the top cover.

If such a cooler was dealing with a gaseous mixture, an outlet for the non-condensable gases would have to be provided at the bottom of the casing as well as the outlet for liquid shown.

Figure 32 shows a construction of a cooler for dealing with a gas at atmospheric pressure. This consists of a set of vertical cooling grids contained in a chamber constructed of sheet lead with burnt joints. The lead is supported from a heavy timber framework, which also serves as part of the insulation construction. The framework is sheathed inside and out with boards, and the intervening space packed with granulated cork. The gas mixture enters at the top and flows downwards. The non-condensable gases and the condensate are removed at the bottom.

Cooler Surfaces.

The heat transmission from pipe surfaces to air or

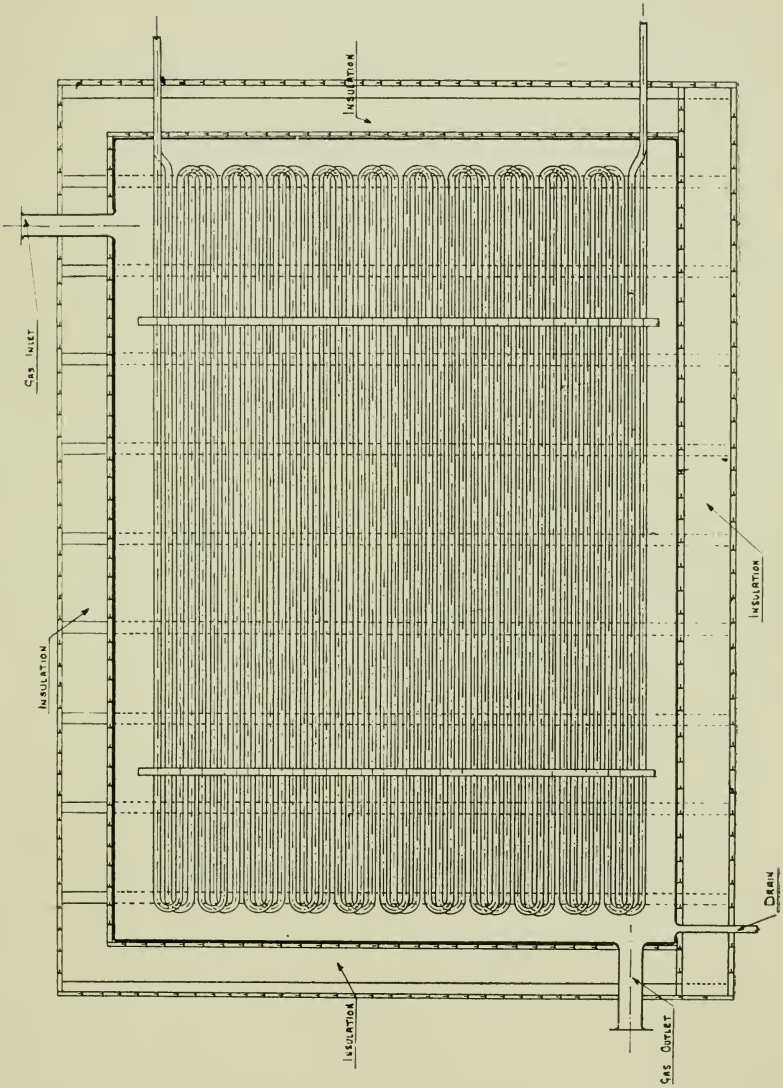

Figure 32

any other gas is very low, ranging fom 3 to 5 B.t.u. per square foot per hour for each degree Fahrenheit difference

in temperature between the inside and outside of the pipe. These rates are for pipes cooling air or gas which is comparatively still, as in an insulated room or gas condenser, such as Figures 29 and 32. In the case of coolers in which a fan maintains a forced circulation the rate may reach 10 or more.

Direct Expansion *v.* Brine Cooling.

In all cases of the cooling of gases, including the cooling of the atmosphere in an insulated room or space, two systems may be used, namely, direct expansion piping or brine piping. In the first case the refrigerant is evaporated directly inside the pipes in contact with the air, while in the second the refrigerant cools brine in a separate tank, this brine being then circulated inside pipes in contact with the air or gas.

In cooling gases the direct expansion system is usually lower in first cost and more economical in operation. If, however, any interruption is caused to the supply of gas being cooled, evaporation of the liquid refrigerant practically ceases, and large amounts of liquid are pulled into the compressor, with the consequent risk of serious damage. This would not happen with the brine system, as the machine would continue to work, cooling the brine, and accumulating a reserve of cold in it.

When cooling rooms, the direct expansion system, while cheap, has the disadvantage of being unable to keep down the temperature of the room when the machinery is stopped. With the brine system the brine stored up in the pipes can absorb a considerable amount of heat, and with proper design an even temperature may be maintained when the machine is stopped, say, over night.

We therefore find the brine system is often adopted, although more costly and not so economical.

Chapter IX.

INSULATION.

All tanks, chambers, and refrigerators in which cooling is done must be effectively insulated to minimize the leakage of heat into the cold parts from the surrounding atmosphere.

Considerable misunderstanding occurs on the subject of insulation for low temperatures. Materials for high temperature insulation and figures for insulating value obtained from steam pipe and similar tests are of no value for low temperature work.

The ideal insulator should contain a large number of small cells, spaces, or interstices containing air. The larger the number and the smaller these spaces, the better. If such spaces are naturally sealed so that the air confined in them cannot circulate about, we have the material for our purpose. The insulating medium should, of course, possess other features, such as not too high first cost, no deleterious action on other materials of construction which come in contact with it, non-hygroscopic, non-combustible, not capable of harbouring vermin, etc.

Many substances have been tried for insulation work, the most largely used being flake charcoal, silicate cotton, or slag wool, and cork, either in the granulated or slab form. Laboratory tests do not indicate any material to be greatly superior to the others. The following figures are taken from tests made by the National Physical Laboratory: —

Granulated Cork ...	0.32	Heat transmission in B.t.u. per square foot of surface per hour per degree F. difference in temperature, the thickness of the insulation being one inch.
Slab Cork	0.33	
Slag Wool	0.30	
Flake Charcoal ...	0.34	

It will be noticed that the rates of heat transmission are generally similar. In practice, however, we find that over a period of time, cork, particularly slab cork, gives us the best results. This is due to the very hygroscopic nature of the other materials. Due to the difference in temperature

across the insulation, there is not only a tendency for the air contained in the insulation to circulate round and round carrying heat with it, but at many points in the insulation some of the cooled air will escape into the atmosphere and a corresponding amount of warm air enter the insulation. This warm air carries with it moisture, which is largely condensed out and absorbed by the insulating material. The insulation becomes wet, loses its insulating value, and the wet insulation causes deterioration of any timber or metal structure which is in contact with it. The great point about insulation work is to use a material which is as little hygroscopic as possible, and to make the insulation as airtight as possible.

Insulating work can be divided into two kinds: —

(1) The insulation of rooms and spaces.
(2) The insulation of vessels, apparatus, etc.

In chemical works practice we shall mostly have to deal with work of the second and most difficult class. It is absolutely impossible to fit an airtight insulation to vessels and yet leave the shell of the vessel accessible for repair, etc. Also there are so many pipe connections which necessarily have to pass through the insulation that the degree of airtightness met with in cold storage practice is out of the question.

Insulation of Vessels.

Figure 33 shows the insulation of a circular tank using loose granulated cork, kept in place by boards. Vertical grounds are wedged between the angle iron flanges round the top and bottom of the tank; these grounds carry timber frames encircling the tank, to which the boards are srcewed or nailed. Wood boxes must be constructed round any pipes connected to the tank, and these are afterwards stuffed with silicate cotton or loose hair felt. This can be packed tightly in place and the box closed by a wooden cover fitting close round the pipes. Such boxes constitute a source of weakness in the insulation, but they are unavoidable.

Figure 34 shows the insulation of a rectangular tank

with two layers of cork slabs. Two layers of cork slabs are laid on the floor in hot asphalt and the tank placed thereon. The whole of the top of the upper layer of slabs should be flooded with asphalt before the tank goes in place. Vertical wood grounds are wedged between the angle iron flange round the top of the tank, and the pro-

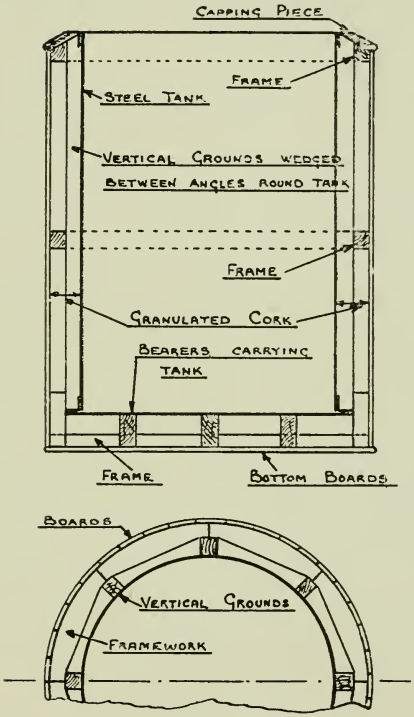

Figure 33

truding edge of the slabs underneath the tank. These grounds are spaced apart a distance equal to the width of the cork slabs used. Slabs are fitted in between the grounds and bedded to the tank side with hot asphalt or Portland cement. The slabs are also nailed to the grounds. A second layer of slabs is placed over the first layer and bedded to it in Portland cement or hot asphalt, and also nailed to the under layer. The outer surface of the insula-

tion may be cased with boards or plastered with a Portland cement mortar. This is a far more effective insulation than the first example, but it is of a very permanent nature, and the tank exterior is not accessible without cutting out the insulation. The frequency with which the outside of the tank will require attention must therefore be considered before deciding the kind of insulation to be fitted, and this frequency will depend upon the nature of the process the tank is employed in.

Reference should also be made again to Figure 32 for a method of insulating a cooling chamber constructed of sheet lead. Cork slab insulation is not advisable in such a con-

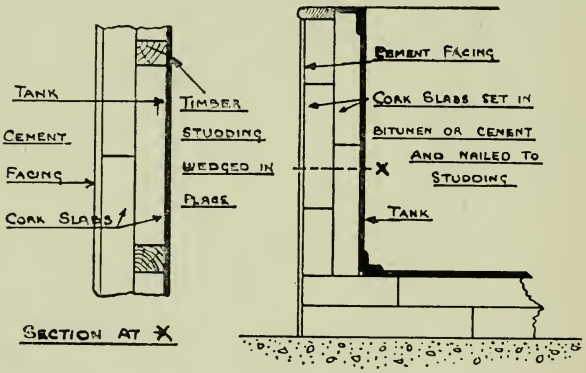

Figure 34

struction because the sheet lead requires renewal at intervals, and this means disturbing the insulation. If it is particularly desired to use cork slab, the lead chambers should be placed in an insulated room and not themselves directly insulated.

Insulation of Rooms.

If there are a number of vessels to be insulated, the best plan is to place them in an insulated room, as shown in Figure 35. This permits of an easy and excellent job being made of the insulation and at the same time leaves the whole of the vessel and pipe connections accessible for examination and repair.

Figure 35 indicates the room insulated with loose cork. If any part of the room is bounded by brick walls or similar existing structure, these are first waterproofed by bitumen or asphalt. Timber grounds on edge are then secured to the wall either by plugging or by securing to runners top and bottom. A double layer of planed, tongued and grooved boards is nailed to the edge of the grounds, waterproof paper being placed between the two layers of boards. The space between the brick wall and the boards is packed with granulated cork.

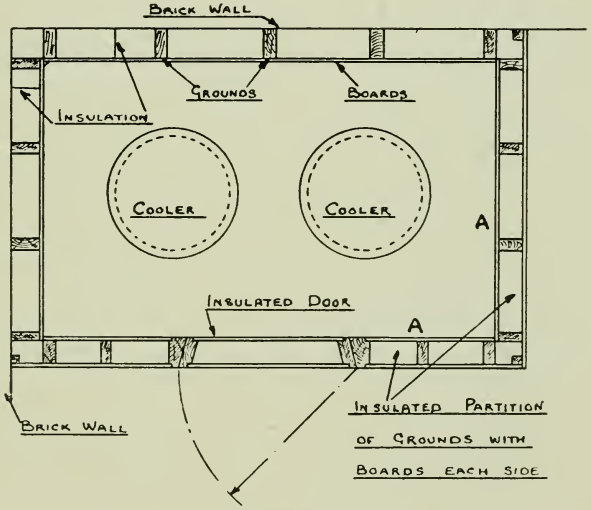

Figure 35

If the walls of the room have to be constructed, as at *A*, Figure 36, they are formed by erecting the vertical grounds and nailing two layers of boards on each edge of the grounds, the intervening space being filled with granulated cork.

Ceilings and floors are insulated similarly. Whatever the nature of the floor insulation, grounds should be provided to take the weight of any heavy tanks or vessels resting on the floor. Floors should also be rendered water-tight by covering with sheet lead or asphalting.

If brick walls have to be insulated with cork slabs, these are simply set up to the walls in hot asphalt or Portland cement mortar. Subsequent layers of slabs would be set up to the first layer in a similar manner and if desired also nailed thereto. The outer surface of the last layer can be finished with a Portland cement plaster, magnesia cement plaster, or matchboarding.

If the wall has to be formed of cork slabs, a framework of grounds is first set up and the first slabs nailed thereto. Additional layers are secured as before and the surfaces plastered or boarded.

An insulated space must have a suitable insulated door, the size of this being sufficient to allow the removal, either wholly or in portions, of the apparatus placed inside the room. Such doors are usually constructed with pitch pine frames with bevelled faces covered with felt. A cross section through such a door and frame is shown in Figure 35.

Insulation of Pipes.

Pipes conveying cold fluids may be insulated with cork sections curved to suit the radius of the pipe. Before applying the insulation the pipes should be cleaned and painted. The cork sections are fitted to the pipes in halves and arranged to break joint with each other. The halves are secured by wiring and the exterior surface finished with a magnesia or similar cement. If a less permanent finish is desired the cork sections may simply be wrapped round with canvas and painted with bitumastic.

Alternatively if there are a number of pipes running alongside one another, a wooden casing may be constructed round them and packed with granulated cork.

Pipe joints and flanges should be left accessible and cased in only with easily removable covers.

Heat Leakage through Insulation.

The amount of heat passing through an insulation depends enormously on the degree of airtightness achieved. In refrigeration work such heat leakage is usually expressed in B.t.u. per square foot per degree F. difference in tempera-

ture per hour (or per 24 hours). Taking the National Physical Laboratory test figures previously quoted we can construct Table 12a, which shows the heat leakage through cork slab of different thicknesses.

Table 12a.

B.t.u. leaking through 1 square foot of cork slab per hour per degree F. difference in temperature.

	Thickness of Cork Slab.					
	1-inch.	2-inch.	3-inch.	4-inch.	5-inch.	6-inch.
Heat leakage ...	0.33	0.155	0.11	0.0775	0.066	0.055

Seeing that the comparative test figures for the other insulating materials are not very different from those for cork slab, Table 12 should also approximately represent the heat leakage through these as well. Such a result, however, only holds providing the other constructions are made and maintained as equally air and moisture tight as the cork slab; a result not generally attained in practice. We therefore find that insulation formed of boards and loose packing needs to be from $1\frac{1}{2}$ to 2 times as thick as cork slab insulation.

For temperatures down to about 30° F. (−1.1° C.) it is usual to use 3-inch thickness of cork slab, increasing to 6-inch thickness for temperatures of 15° F. (−9.5 C.).

Chapter X.

OPERATION OF REFRIGERATING PLANT.

With regard to the starting and stopping of refrigerating plants, instructions for this will always be given by the manufacturers of the particular plant, and while the general sequence of operations is always the same, there may be variations in minor instructions depending upon the type of plant installed.

In general, at starting, the water service to the condenser should first be turned on and all valves between the condenser and compressor, including the by-pass, opened. Lubricators being filled and adjusted the machine may be started and run up to speed. The by-pass would then be closed and the main suction stop valve gradually opened. This should never be opened suddenly because accumulations of liquid refrigerant may be lying in the pipes between the refrigerator and the compressor, and these, if suddenly drawn into the compressor, can cause serious damage. When the suction stop valve is fully open, the regulating valve is then gradually opened and adjusted as required.

In stopping the reverse process is gone through, the regulating valve being first closed and any excess refrigerant pumped out of the refrigerator. The suction stop valve will then be closed and the machine stopped. Finally the valves on the delivery line will be closed and the water shut off the condenser. Adherence to a regular routine will prevent accidents due to improper operation.

CO_2 machines are frequently stopped with all valves left open and the pressures allowed to equalise all round the system. This is especially done if the stoppage is only of short duration.

In the case of a heavy inrush of liquid refrigerant into the compressor the first step to take is to stop the machine, or if steam driven, it may be reduced to a dead slow speed. With electrically driven machines such an inrush of liquid may itself stop the machine by tripping the circuit breakers.

The suction stop valve should then be almost closed, and the regulating valve shut. The machine should then be run, at a slow speed, if possible, to work the excess liquid over gradually into the condenser. As this process proceeds the suction stop valve may be slowly opened wider and wider until it is well open and the refrigerator pressure has fallen somewhat below its normal value when the usual running conditions may be reverted to. The cause of the excess liquid should be investigated and remedied. With direct expansion gas coolers it may be due to interruption of the gas supply to the cooler. Or it may be due to the regulating valve being too much open in other cases.

The machine should be kept well charged with refrigerant. The sufficiency of the charge can in general be ascertained from the condenser pressure, which should be that corresponding to a temperature some 10° F. (5.5° C.) above that of the condenser water supply. An overcharged machine will show an excessive condenser pressure, and the condenser pressure gauge will be jumpy.

The presence of foreign gases (air, etc.) will also cause an excessive condenser pressure; as will also an insufficient or improperly distributed water supply to condenser coils.

An undercharged machine will have too low a condensing pressure; will require the regulating valve to be opened much wider than usual, and will not be able to keep its temperatures.

When it becomes necessary to add more refrigerant to the system, great care should be taken to see that this is pure and anhydrous. If it contains any moisture, trouble will be subsequently experienced in the working of the plant and its capacity will fall off. CO_2 machines are more susceptible to moisture trouble than ammonia. The latter substance dissolves in the moisture forming a liquor which does not freeze so readily as the moisture alone. Where there is any doubt of the dryness of the refrigerant it should be passed through a dryer. CO_2 may be dried over calcium chloride, the outlet from the dryer having a filtering pad of cotton wool to prevent small particles of calcium and

dust being drawn into the compressor. CO_2 cylinders may also be stood upside down for 24 hours before emptying. If the valve is then very gently opened a little while in this inverted position the majority of the water present can be drained out. Ammonia may be dried over quick lime. In some CO_2 machines a fitting is arranged through which methylated spirit can be added to the system to prevent the freezing of any water present.

The clearance in the compressor cylinders being very small, it is necessary to see that this is properly divided between the two ends whenever the machine has been overhauled. This is done by taking out a valve at each end of the compressor and inserting pieces of lead wire. If the compressor is turned through a revolution, the wires will be squeezed between the piston and cylinder cover, thus indicating the exact amount of clearance.

A complete set of spare valves and seats should be kept and changed at regular intervals. The set which has been working can then be faced and ground at leisure ready for replacement when the time for the next change arrives.

It is advisable to introduce a regular routine with regard to the cleaning and overhaul of the plant, even to the cleaning of the electric light globes. The intervals of time for such operations must be determined by experience in the individual case, but once so determined regularity should be insisted upon.

Most plants suffer in that they are not equipped completely with measuring instruments, *e.g.*, thermometers, etc. In fact, many plants have only the condenser and refrigerator pressure gauges, any other information desired having to be obtained by inference. Means should be available for ascertaining the temperatures of the condensing water, of the brine (if used), of the substance to be cooled both entering and leaving the cooler, and of the compressor suction and delivery. Machines can be worked without such information, but the control of a large plant can be effected infinitely better with its aid. Quantitative measurements should also be made where possible, *e.g.*, power used, amount of brine or other substance cooled,

and so on. A proper log sheet should be drawn up and the information recorded thereon at regular intervals, by the man in charge. Such a course has the further advantage that it makes the operator go all round the plant regularly, and thus there is no excuse for any irregularity escaping attention for long. For example, with condensers placed on the roof these may have a quite sufficient supply of water and yet not be working to advantage owing to unequal distribution over the coils. If the operator has to mount to the top of the condenser regularly to ascertain the temperature of the water supply, such an unequal distribution is brought to his notice as soon as it occurs, and it can be remedied. Such temperatures can be conveniently taken by a system of long distance electrical thermometers. These, however, do not possess that advantage of ordinary thermometers, namely, that the operator must actually visit all portions of the plant where thermometers are situated, and thus has defects brought to his notice. In a large plant ordinary thermometers and instruments should be installed for use of the operators, with an electrical system for use of the superintendent of the plant.

With large plants it is the author's practice to average such log sheets weekly, taking out a balance sheet for the work done, contrasted with that which should have been done. Any falling off in capacity can thus be at once detected.

We have already discussed the systems of working known as wet and dry compression, and pointed out that the majority of machines work on a combination of both systems. Under such conditions a thermometer placed in the compressor suction would record a temperature corresponding to the pressure in the refrigerator. Many tests of machines, both large and small, working on ammonia and carbon dioxide, have shown in the author's experience that under such conditions the output of the machine is a maximum and the working most satisfactory. We then find that the temperature of the gas discharged from the compressor is about 140° to 180° F. (60° to 82° C.). The average ammonia machine would incline to the lower tem-

perature and the CO_2 machine to the upper temperature. In tropical situations, where the condensing water is warm, much higher temperatures may be attained. In such cases we find that there is a continual accumulation of foreign gases in the system, which require to be periodically blown out if good working is to be ensured. Such gases may arise partly from the lubricating oil and from some decomposition of the refrigerant. Each blowing out or purging carries away a little of the charge, which must ultimately be made up again. Such blowing out should be done after the plant has been standing some hours, or over a week end. The foreign gases will be found to have accumulated in the top of the condenser, and can be readily blown off by the valve placed on top of the condenser for this purpose. It is usual to blow the gases from ammonia machines into a bucket or vessel of water, this serving the double purpose of preventing any smell of ammonia, and indicating when all the foreign gases have gone. So long as non-condensable gases are escaping, bubbles rise through the water, but when ammonia begins to come off, the bubbles cease and a crackling noise ensues, due to the ammonia bubbles dissolving in the water. If desired, a similar method can be used for CO_2, substituting caustic soda instead of water. CO_2 machines are not troubled to anything like the same extent as ammonia machines with such accumulations.

The lubricating oil used in the compressors must be chosen with discretion. The oil must not be such as to saponify when used with ammonia; it must not unduly thicken or solidify at the lowest temperature met with in the system; it must not be so light an oil as to carbonize at the temperature of discharge from the compressor. Mineral oils specially for use with refrigerating plant are obtainable and should be used. The oil recovered from oil separators can be used over again after being filtered.

In plants using brine this is usually made from calcium chloride, and not from sodium chloride. Calcium chloride does not attack iron pipes, provided they are kept submerged below the surface of the brine. If the pipe surfaces are

alternately wet and dry, corrosion may go on rapidly. Sodium chloride brine does attack iron pipes, but it is said to be rendered innocuous if it is made definitely alkaline. Brine pumps not in use should be moved frequently or their spindles and rods are liable to set fast. In some cases brine made of magnesium chloride is used. Tables 13, 14, and 15 gives the properties of brines of different strengths.

Chapter XI.

CHOICE OF A PRIME MOVER.

Refrigerating machines may be operated by any kind of prime mover, of course, but different kinds of prime mover have different characteristics, which must be borne in mind when installing a plant.

Steam Drive.

Formerly the machines were steam driven, either through a belt or directly coupled. The steam engine possessed three features of great service in the operation of refrigerating machines, namely—

(1) power to start the machine from rest against the full load.
(2) large overload capacity.
(3) great flexibility as regards speed of operation.

Unfortunately, however, the steam engine has numerous disadvantages, such as lack of economy, excessive floor space occupied, especially when the boiler plant is taken into consideration, etc., so that we do not now find many steam driven machines outside of marine practice and works which already have boiler plant for other purposes, and works in remote situations. Extensive use of the steam drive is still to be found in U.S.A., although even there it is losing ground.

Electric Motors.

The most popular prime mover at present is the electric motor, which occupies a minimum of space, is always ready for immediate work without preparation, and yet requires no attention when not in use.

The direct current motor will start the machine against its load, and has a definite but limited overload capacity. Under the present British standard rating conditions motors are rated for continuous operation at the maximum load

they can operate at continuously, this maximum load including an allowance for an overload. Such motors have some range of speed variation, usually about 10 per cent. of the normal speed. A greater range can be obtained by special arrangements at greater cost. All direct current motors fitted for refrigerating machine operation are of the compound wound type.

Some designs of alternating current motors have not sufficient starting torque to start the machine from rest against full load. A bypass is therefore fitted to release the load until the machine comes up to speed. Except in certain types of alternating current motor there is no speed variation, and we cannot adjust the speed of the refrigerating machine to suit the work asked from it. The machine must be running at full speed, or not at all. In some cases this may constitute a great disadvantage unless great care is taken to proportion the size of the plant to the work required of it. If the refrigerating machine is too big for its work, intermittent operation will be necessary, and this may cause an objectionable cyclical variation in temperature, especially with a direct expansion system.

In works which already have a number of alternating current motors driving other plant, the possibility of using synchronous motors for the refrigerating machine should not be overlooked, as these can be arranged to give an improved power factor to the electrical system, and often secure lower power rates.

A further point to receive consideration with the electric drive is the possibility of an interruption in the source of power, when the power is drawn from an outside source. A sympathetic or lightning strike of power station operatives may easily put the refrigerating machine out of action when it is most required.

Internal Combustion Engines.

Any type of internal combustion engine can be used for driving refrigerating machines. None can start the machine from rest against its load, and bypasses must always be fitted. The method of starting should be sufficiently

powerful to start the engine and bypassed compressor together without having recourse to fast and loose pulleys. In fact, no fast and loose pulleys should be permitted on any belt-driven machine, except in the smallest sizes.

The operation of refrigerating machines is arduous work, involving long periods of continuous running. The prime mover should therefore be installed with a margin of power. This is especially necessary when the driving is done by internal combustion engines, which usually have no overload capacity. The full load imposed by the refrigerating machine should not then exceed 80 per cent. of the normal capacity of the internal combustion engine.

In deciding the power of the prime mover it must also be remembered that the power taken by the refrigerating machine will depend upon the temperature conditions in the condenser and refrigerator, and will be a maximum in hot weather. Due consideration must therefore be given to possibilities of variation in working conditions, and a margin of power provided to meet them. The tables of mean pressure given in Chapter 3 will indicate what is necessary in this respect.

When the prime mover and refrigerating machine run at different speeds they may be coupled by belts, ropes, chain, or toothed gear. The two latter must be designed by specialists for satisfactory results. Belts are suitable for small machines, while cotton ropes give excellent service on larger ones. The best and modern practice is, however, to use high speed machines directly coupled to the prime mover, thus saving in first cost and space, and gaining in efficiency.

Foundations should be of liberal size, so that there is no chance of any settlement disturbing the alignment of the machine. Satisfactory running is largely a matter of preserving the alignment which the machine has when first made.

Chapter XII.

CHEMICAL PROCESSES USING REFRIGERATION.

The applications of refrigeration in the chemical industry and allied industries are so extensive that it would be impossible to notice them all within a reasonable compass. The following notes, however, include a representative collection of such uses, either actual or proposed, and the author will be glad to have any important omissions brought to his notice. It should be noticed that in many cases whether refrigeration is or should be applied to a process will depend on the situation in which the particular process is intended to be carried on. Thus in England many processes can be successfully worked with the cooling operations done by natural supplies of cold water, perhaps supplemented in the summer months by refrigeration. The same processes carried on in U.S.A. or Australia would certainly require artificial cold.

Air Liquefaction.

The liquid air industry is now of great importance as it yields enormous quantities of oxygen and nitrogen. While not a strictly chemical process, its importance warrants mention here. Apart from the actual liquefaction being brought about by artificial cold, refrigeration is used to dehumidify the air before liquefaction, this being essential to the satisfactory operation of the liquefying plant.

Alcohol.

Refrigeration is stated to play a part in the purification of alcohol.

Ammonia (Synthetic).

Refrigeration is mentioned as a part of the Haber process for the synthetic production of ammonia. After the combination of the nitrogen and hydrogen has been effected the gases containing the ammonia are stated to be

cooled below −65° F. (−60° C.) to effect condensation of the ammonia. Such temperature is necessary on account of the low concentration of ammonia present in the gas mixture. While there is no difficulty in producing the low temperature stated, it is doubtful if such an application in the Haber process has been made outside of the laboratory. Actually the gas mixture is passed through water, which dissolves out the ammonia.

Benzene Chlorination.

It has been suggested that refrigeration could profitably be used during the process of chlorinating benzene. The heat generated is usually carried off by cooling water, and, apart from the fact of this not being at times obtainable cool enough, lower temperatures may be advantageous.

Benzol Recovery.

Refrigeration has been employed in the recovery of benzol from coke oven gases. An installation for this purpose was put down at the Bethune Mines, North France. Coke oven gas, after extraction of the ammonia, is washed with tar oil to remove the benzol. The tar oil is then refrigerated so that it will retain the benzol until the latter can be recovered by distillation. The oil is cooled to 112° F. (44.4° C.) by water, and then cooled to 59° F. (15° C.) by the refrigerating plant.

Bi-carbonate of Sodium.

The use of refrigeration has been proposed in the production of this substance by the ammonia soda process. Ammonia and carbon dioxide are to be used under pressure, say 100 atmospheres, or in the liquid form. The cold produced by their expansion is to be utilized to cool the liquors in the absorption and carbonating towers.

Bleaching.

The production of bleaching liquor by the electrolysis of brine solution, cooling of the electrolyte is necessary to prevent decomposition of the sodium hypochlorite.

Bleaching Powder.

In some processes of manufacture of bleaching powder it is found that in hot weather the bleach is liable to heat up after packing. This necessitates unpacking the casks and spreading the powder on the floor to cool off. All such trouble can be avoided by refrigerating the chlorine gas before absorption by the lime.

Bread.

Refrigeration is also employed in the bakery. It maintains constant temperature and humidity in the dough room, thus preventing faulty fermentation. The process is thus under complete control all the year round.

It is also used for preserving the yeast and the perishable goods, meat, fruit, etc., used for pies and cakes.

Brewing.

Very extensive use of refrigeration is made by the brewing industry to cool the fermenting rooms and cellars, for cooling water for attemperators, for wort cooling, for hop and malt storage rooms, for carbonating, and for chilling beers in order to clarify them.

The cooling of the fermenting rooms gives a purer and drier atmosphere, keeps down the concentration of CO_2 in the room, and prevents any growths of fungus.

Cooling of the beer on its way from the fermenting room to the storage tubs effects precipitation of albuminoids, which can be removed by filtration. Also the cold beer can take up more CO_2 during carbonating.

In some instances chip casks also are equipped with refrigerating coils to effect precipitation of impurities.

Candles.

Refrigeration has been employed to cool the water used for solidifying the candles when poured into the moulds.

It has also been used in the bleaching of candles, its part being to dehumidify air, which was then ozonised and used to effect the bleaching.

Celluloid.

In the manufacture of celluloid refrigeration is used to effect complete recovery of valuable volatile solvent vapours.

Chlorates.

Refrigeration played a part in the old process for the manufacture of potassium chlorate. Potassium chloride was added to a previously prepared calcium chlorate solution, when potassium chlorate was formed and crystallized out. Further cooling of the mother liquors with refrigerating plant gave a further quantity of potassium chlorate.

Chlorine (Liquid).

Refrigeration is used in the liquefaction of chlorine. If the chlorine is condensed under pressure the coils containing the gas are cooled by a brine bath. If the chlorine is condensed under atmospheric pressure, the cooling is done by the evaporation of CO_2 in pipes directly in contact with the chlorine. The trade in liquid chlorine has increased enormously since the beginning of the European War, 1914.

Chlorides, Olefine.

In a proposed process for the production of olefine chlorides a mixture of chlorine and olefines is alternately heated to bring about the formation of the chlorides and cooled to about $-4°$ F. ($-20°$ C.) to separate out the chlorides.

Dyes, Aniline.

Artificial cooling is made use of in the crystallizing out of salts from solutions. In this class of work the refrigerating machine appears to be usually employed in making ice which afterwards does the cooling. The temperatures must be controlled during the process of manufacture, or the dyes decompose.

Dyeing.

It is stated that in the dyeing and printing of fabrics with aniline colours, the fabric is first given an alkaline

base with napthol and dried at a low tempeature produced by refrigerating machinery.

Again, in the dyeing of silks using bi-chloride of tin, it is stated that considerable loss was occasioned, due to the precipitation of particles of tin on the silk fibres at atmospheric temperatures. This has been obviated by artificial cooling.

Distilling.

Refrigeration is sometimes used in the distillery to cool the mash in the fermenters, and for keeping the yeast at the right temperature.

Electric Batteries.

Artificial cooling of electric storage batteries has been found advisable, especially where a maximum of capacity is desired with a minimum weight and size of battery. Probably the first practical application of refrigeration for this purpose was in a plant designed for a submarine in 1913.

Explosives.

In the manufacture of high explosives refrigeration plays a very important part. Control of the temperature during the nitrating process is essential, and without refrigeration the manufacture could not proceed with certainty and safety.

Again, in processes in which guncotton is treated with ether and alcohol, pressed, and the excess alcohol and ether driven off, refrigeration is made use of to recover the alcohol and ether.

Time fuses, as for shrapnel shells, are made up of a very hygroscopic powder. At the same time great certainty and nicety of action are insisted upon. The action of the fuse is considerably affected by its moisture content, so that the humidity of the atmosphere must be controlled and kept constant to within 2 per cent. during the manufacture and packing of the fuse. The humidity is controlled by refrigeration, as described in Chapter 8.

An ever-increasing use is being made of refrigeration generally to control atmospheric conditions in many industries, especially during the manufacture and storage of explosives.

All naval ships now have refrigerating plant to cool the magazines and prolong the life of the ammunition.

Glue.

After boiling and filtering, the mass is run into congealing tanks to set. The quality depends upon temperature control, for which refrigerating machinery is now largely used.

Glaubers Salt.

This is recovered from the salt dumps of the German potash industry. The magnesium sulphate and salt solution is treated in concrete tanks provided with agitating gear and iron cooling coils through which cold brine is circulated. The Glaubers salt separates out as fine crystals. Large crystals can be produced by cooling very slowly and avoiding all disturbance of the crystallizing vat.

Gum.

Refrigeration is used in the manufacture of gum.

Hydrochloric Acid.

It is stated that refrigeration is used in the process of purifying hydrochloric acid, being employed to effect the separation of the arsenic compounds.

Ink.

Refrigeration is used in the manufacture of coloured inks in which intermediate coal tar products are used.

Margarine.

Refrigeration takes its part in the manufacture of margarine, being used in the kernelizing or graining out of the product from the rich mother emulsion. In some cases the mixture of hydrogenated and normal oils has been passed through coils immersed in a cooling tank, with the result

that vegetable stearine was liable to separate out and block the coils. The oil mixtures are better run over chill rolls cooled to 5° F. or 10° F. (−15° C. to −12.2° C.).

Meat Offal By-products.

Refrigeration is used in the recovery of stearine and similar substances from the meat offal in meat packing houses.

Medicines.

The number of pharmaceutical products in the manufacture of which refrigeration plays some part is too large to permit of individual mention. One of the latest discoveries, insulin, makes considerable use of refrigeration. This is prepared from the pancreatic glands of mammalian animals, the process involving a large number of operations. Refrigeration is employed in the cooling of the raw material, which is ground or mixed at a low temperature, in the recovery of alcohol with which the mixture is treated, in the clarifying of the alcohol extract, and for the concentration at a low temperature.

Mercerising.

In the mercerising process the fabric is immersed in a bath of caustic potash or soda lye, then washed in water, in dilute sulphuric acid, in water, and again in soda solution. The soda lye is stored at about 40° F. to 44° F. (4.4° C. to 6.6° C.) and chilled from 32° F. to 39° F. (0° C. to 3.9° C.) for the mercerising process.

Oils, Fish.

Refrigeration is used to separate out stearine from fish oils.

Oils, Vegetable.

Oils produced from cotton seed, linseed, rape, castor beans, niger seed, cocoanuts, copra, sesame, olives, etc., may also be refrigerated during some part of the process of purification and the precipitation of the stearine, etc.

Oils, Mineral.

Refrigeration is made use of in the recovery of volatile hydrocarbons from gases escaping from oil wells. In one method the gases are compressed and then allowed to expand, thus reducing their own temperature. Where absorption of the volatile products is effected by washing with an oil, such oil is cooled by the refrigerating machine.

Refrigeration is also used in the distillation of mineral oils. From the first distillation products, paraffins are separated out at a low temperature as fine crystals. Later, less volatile and softer paraffins, which will dissolve very readily in the mother lye if the temperature rises, are cooled by refrigeration so that quick separation is effected and then filtered out.

The use of refrigeration during such distillation has given us the uniform and satisfactory lubricating oils at present attainable. The lighter oils may be chilled to from 10° F. to −10° F. (−12.2° C. to −23.3° C.) to remove the wax. The chillers may take the form of jacketed vessels with agitating gear. The cooled oil is then pumped through filter presses.

Heavy oils are sometimes cooled by storage in tanks in a refrigerated room.

Artificial cooling may also be resorted to in the racking or barrelling of paraffin during hot weather.

Paper.

Refrigeration is used for temperature control in the sulphite fibre works where the wood pulp is transformed into paper. It is used again to control the temperature of the acids used in the preparation of parchment paper. It is also used in the making of waxed paper. The paper runs over a hot roll dipped into melted paraffin and then over a chill roll. The latter may need refrigeration in hot weather.

Perborate, Sodium.

A coarse crystalline and stable form is produced by dissolving borax and sodium hydroxide in boiling water

and adding the solution, cooled to 60° C., to a solution of hydrogen peroxide. The mixture is stirred and cooled below 10° C. and allowed to crystallize slowly.

Percarbonate, Potassium.

This is prepared by the electrolysis of a saturated solution of potassium carbonate at a temperature of 14° F. to 3.2° F. (−10° C. to −16° C.).

Percarbonate, Sodium.

This is made by adding sodium peroxide slowly to ice cold absolute alcohol. Dry CO_2 is passed into the mixture at 32° F. to 41° F. (0° C. to 5° C.).

Perchlorates.

Refrigeration is used in the preparation of sodium perchlorate by electrolysis. A weak solution of sodium chloride is electrolized, producing perchlorate with some chlorate. This chlorate is dissolved in water and separately electrolized. The bath is cooled by refrigeration plant or the electrodes themselves may be hollow and also artificially cooled.

One point must be carefully noted in applying refrigeration to the cooling of such liquids during electrolysis. The liquid forms part of the electric circuit and the introduction of the cooling apparatus must not be made in such a way that it provides a leakage to earth for the current.

Perfumes.

Artificial cooling is required in the manufacture of perfumery. It is used when extracting the aromatic principle from flowers, and also during the processes of distillation, rectification, and separation of the various essences. It also preserves the flowers and animal perfumes until they are required, and it is used in the manufacture of synthetic perfumes.

Peroxide, Hydrogen.

Refrigeration is stated to be used in the production of a pure aqueous solution of hydrogen peroxide from pure hydrated barium peroxide and dilute sulphuric acid.

Peroxide, Calcium.

One method of preparation involves the addition of sodium peroxide and slaked lime in molecular proportions to iced water. Refrigeration, of course, produces the latter.

Peroxide, Magnesium.

This may be prepared by adding sodium peroxide and alcohol to aqueous magnesium chloride. Magnesium peroxide is precipitated and further cooling by refrigeration yields considerable additional magnesium peroxide.

Persulphuric Acid.

This may be produced by the electrolysis of sulphuric acid at low temperatures.

Persulphate, Ammonium.

This may be prepared by the electrolysis of a saturated solution of ammonium sulphate at a low temperature.

Photographic Materials.

Refrigeration is very important in the manufacture of photographic plates, films, etc., being used to control the temperature and humidity of and remove dust from the atmosphere.

Polish, Shoe.

It is stated that a number of shoe polish manufacturers in U.S.A. employ refrigeration in their processes.

Rubber.

Refrigeration has long been an ally to the rubber industry. It is said to be used right from the start, the sap being cooled at the time of its collection. In the manufacture of rubber products from the crude material, the temperature and humidity must be controlled. In the cutting of thin rubber strip from blocks both the blocks and the cutting knives may require refrigeration during warm weather. The humidity in the drying room for motor tyres is also controlled by refrigeration.

Sugar and Molasses.

In these processes also refrigeration is used to effect crystallization. Saccharine juices are concentrated by freezing and decantation. Raw materials may be stored in refrigerated rooms. In the production of sugar from molasses, decomposition and separation of the saccharates are facilitated by cooling. Refrigeration is also used in the liming process.

Solvents, Volatile.

In many processes employing volatile solvents, as in the manufacture of artificial silk, refrigeration may be used to effect complete condensation and recovery of the valuable solvent vapours which would otherwise be lost.

Soap.

Refrigeration is extensively used in the soap and washing powder factories in U.S.A.

Tanning.

The American tanning industry employs refrigeration to cool the tanning liquor and colouring matter and lengthen its effective life.

Tar Distillation.

In a method of manufacture of lubricants from the products of crude tar distillation, refrigeration is used to produce a better product more free from suspended matter by freezing out the phenols and de-paraffining at about 14° F. (−10° C.).

Tobacco.

If tobacco leaves are cured on the fermenting process, temperature control is essential or the aroma may be lost. If the non-fermenting process is used, the temperature and humidity must be cooled to prevent sweating. Low temperatures reduce the loss by tobacco weavil during storage of the leaf and refrigeration is also used to dehydrate and cool the air in the storage rooms for snuff.

Tea.

In the preparation of tea, the leaf goes through a fermentation process in which control of the temperature is necessary. This, it is stated, is now better accomplished by refrigeration than by relying on natural supplies of cooling water.

Viscose.

In the manufacture of viscose it has been proposed to use refrigeration to cool the caustic alkali solution to about 28.4° F. (−2° C.) before adding the cellulose.

Wine.

Refrigeration is now used in the wine industry for the control of the fermentation process, for concentration, during the ageing process, and for clarification.

Yeast.

Manufacturers of yeast use refrigeration to control the temperature of the mash and culture of the yeast body, and for the preservation of the finished product.

Gelatine.

Considerable use has been made of refrigeration for cooling in gelatine works.

Gutta Percha.

The manufacturers of gutta percha use refrigeration.

Linoleum.

Refrigeration has found application in the manufacture of linoleum.

Matches.

It has also been used in the manufacture of matches.

Metallurgy.

Refrigeration found an application in metallurgy in the dehumidification by cooling of the air supplied to blast furnaces. The dehumidified air containing only a small

and constant moisture content gives more uniform working of the furnace, a better product with less fuel consumption and a reduced expenditure of power for the blowing engines. Some very large plants have been erected for this purpose, but in one or two instances their use has been abandoned.

Some use of cold air has also been made for the cooling and hardening special steel products such as tools, surgical instruments.

APPENDIX.

The following are some freezing mixtures for laboratory use, and the temperatures attainable with them.

Materials.	Proportions.	Temperatures.
Snow or Ice Sodium Chloride	2 1	−5° F (−20.5° C)
Ammonium Nitrate Water	1 1	4° F (−15.5° C)
Sodium Sulphate Ammonium Nitrate Dilute Nitric Acid	6 5 4	−40° F (−40° C)

Properties of Sodium Chloride Brine.

Table 12.

Specific Gravity	Freezing Point °F	Freezing Point °C	% Na Cl by weight	lb. Na Cl per gallon	lb. Na Cl per cu. foot	Weight of 1 gall. lb.	Specific Heat
1.076	18	−8	10	1.076	6.73	10.76	.892
1.115	11	−11.7	15	1.67	10.43	11.15	.855
1.155	0	−17.8	20	2.31	14.5	11.55	.829
1.183	−8	−22	23.5	2.78	17.4	11.83	.800

A 23.5 per cent. solution of NaCl is the strongest that can be used in refrigeration work with advantage. A denser brine than this would still freeze at the same temperature of −8° F.

Properties of Calcium Chloride Brine.

Table 13.

Specific Gravity	Freezing Point °F	Freezing Point °C	% $CaCl_2$ by weight	lb. $CaCl_2$ per gallon	lb. $CaCl_2$ per cu. foot	Weight of 1 gall. lb.
1.12	15.8	−9	15.5	1.736	10.85	11.2
1.14	8.6	−13	17	1.938	12.1	11.4
1.16	3.2	−16	19	2.205	13.75	11.6
1.18	−4	−20	21	2.475	15.47	11.8
1.20	−11.3	−24	23	2.76	17.25	12.0
1.22	−20.2	−29	25	3.05	19.05	12.2
1.24	−29.2	−34	27	3.35	20.9	12.4
1.26	−40	−40	29	3.65	22.8	12.6

The following table gives the specific heat of calcium chloride brines of various strengths. It is taken from the Report of the U.S.A. Bureau of Standards presented to the American Society of Refrigerating Engineers in 1921, and is the most accurate data available on the subject.

Table 14.

Temperature °F	Specific Gravities			
	1.175	1.200	1.225	1.250
−10			.670	.654
0	.722	.697	.676	.659
10	.728	.703	.681	.663
20	.733	.708	.685	.667
30	.736	.711	.689	.670
40	.740	.715	.693	.674
50	.743	.719	.697	.677
60	.746	.722	.700	.680
70	.750	.726	.704	.684

The densities of brines in refrigeration work are usually measured with a Twaddle hydrometer. The readings of this hydrometer may be converted to specific gravities as follows: —

multiply Twaddle reading by 5 and add 1,000.

thus 40 Tw. = (40 × 5) + 1,000 = 1,200 specific gravity = 12 lb. per gall.

Magnesium Chloride Brine.

A magnesium chloride brine of 25 per cent. strength will freeze at −22° F. (−30° C.).

Hygrometric Tables.

Temperatures		Weight of Water Vapour		Maximum Tension of Aqueous Vapour	
°C	°F	Gms. per cu. metre	lb. per cu. ft.	mm. Hg.	Atmospheres
50	122	82.3	.005142	92.05	0.121
49	120.2	78.62	.004921	87.62	0.1153
48	118.4	74.94	.0047	83.2	0.1095
47	116.6	71.25	.00448	79.15	0.1042
46	114.8	68.1	.00426	75.1	0.0988

Temperatures		Weight of Water Vapour		Maximum Tension of Aqueous Vapour	
°C	°F	Gms. per cu. metre	lb. per cu. ft.	mm. Hg.	Atmospheres
45	113	64.95	.004072	70.45	0.094
44	111.2	61.8	.003863	67.8	0.0892
43	109.4	58.9	.003677	64.4	0.0853
42	107.6	56	.00349	61	0.0803
41	105.8	53.37	.003329	57.95	0.0763
40	104	50.73	.003168	54.9	0.0722
39	102.2	48.22	.00302	52.1	0.0686
38	100.4	45.91	.002871	49.3	0.0649
37	98.6	43.65	.002727	46.75	0.0615
36	96.8	41.39	.002583	44.2	0.0581
35	95	39.35	.002455	41.94	0.0551
34	93.2	37.29	.002326	39.57	0.0521
33	91.4	35.42	.00226	37.47	0.0493
32	89.6	33.55	.002093	35.36	0.0465
31	87.8	31.84	.001987	33.46	0.044
30	86	30.13	.00188	31.55	0.0415
29	84.2	29.58	.001783	29.83	0.0393
28	82.4	27.02	.001686	28.1	0.037
27	80.6	25.6	.001597	26.54	0.0349
26	78.8	24.17	.001508	24.99	0.0328
25	77	22.9	.001429	23.59	0.031
24	75.2	21.62	.001349	22.18	0.0292
23	73.4	20.46	.001276	20.92	0.0275
22	71.6	19.29	.001203	19.66	0.0258
21	69.8	18.24	.001138	18.53	0.0244
20	68	17.18	.001072	17.39	0.0229
19	66.2	16.23	.001012	16.38	0.0216
18	64.4	15.27	.0009528	15.36	0.0202
17	62.6	14.41	.0008993	14.45	0.019
16	60.8	13.55	.0008458	13.54	0.0178
15	59	12.73	.0007975	12.73	0.01675
14	57.2	12.01	.0007492	11.91	0.0157
13	55.4	11.32	.0007059	11.19	0.01475
12	53.6	10.62	.0006626	10.46	0.0138
11	51.8	9.995	.0006237	9.815	0.01295
10	50	9.37	.0005848	9.17	0.0121
9	48.2	8.815	.00055	8.595	0.0113
8	46.4	8.26	.0005152	8.02	0.0105
7	44.6	7.76	.0004841	7.555	0.0098
6	42.8	7.26	.000453	7.09	0.0092
5	41	6.865	.0004253	6.595	0.0086
4	39.2	6.37	.0003975	6.1	0.008
3	37.4	5.975	.0003729	5.7	0.0075
2	35.6	5.58	.0003482	5.3	0.0070
1	33.8	5.23	.0003263	4.95	0.00655
0	32	4.88	.0003043	4.6	0.0061

Temperatures		Weight of Water Vapour		Tension of Aqueous Vapour	
°C	°F	Gms. per cu. metre	lb. per cu. ft.	mm. Hg.	Atmospheres
−1	30.2	4.535	.0002835	4.27	0.00565
−2	28.4	4.21	.0002626	3.94	0.0052
−3	26.6	3.915	.0002444	3.655	0.0048
−4	24.8	3.62	.0002261	3.37	0.0044
−5	23	3.37	.0002103	3.125	0.00405
−6	21.1	3.12	.0001945	2.88	0.0037
−7	19.4	2.9	.0001809	2.67	0.00345
−8	17.6	2.68	.0001672	2.46	0.0032
−9	15.8	2.49	.0001555	2.275	0.0029
−10	14	2.3	.0001437	2.09	0.0026
−11	12.2	2.165	.0001351	1.91	0.0025
−12	10.4	2.03	.0001265	1.83	0.0024
−13	8.6	1.88	.00008605	1.69	0.0022
−14	6.8	1.73	.0000456	1.55	0.002
−15	5.0	1.6	.00003755	1.43	0.00185
−16	3.2	1.47	.0000295	1.31	0.0017
−17	1.4	1.37	.00002305	1.215	0.0016
−18	−0.4	1.27	.0000166	1.12	0.0015
−19	−2.2	1.115	.0000106	1.025	0.00135
−20	−4.0	1.06	.0000066	0.93	0.0012

Dew Points of Air with Varying Moisture.

Temperatures		Rel. Hum. 90 %		Rel. Hum. 80 %		Rel. Hum. 70 %		Rel. Hum. 60 %		Rel. Hum. 50 %	
°C	°F	°C	°F	°C	°F	°C	°F	°C	°F	°C	°F
0	32.0	−1.5	29.3	−3.0	26.6	−4.9	23.2	−6.5	20.3	−9.2	15.4
2	35.6	0.9	33.6	−0.9	30.4	−2.5	27.5	−4.8	23.4	−7.1	19.2
4	39.2	2.4	36.3	0.9	33.6	−0.9	30.4	−2.9	26.8	−5.3	22.5
6	42.8	4.5	40.1	2.9	37.2	0.9	33.6	−1.3	29.7	−3.7	25.3
8	46.4	6.4	43.5	4.5	40.1	2.7	36.9	0.6	33.1	−1.9	28.6
10	50.0	8.5	47.3	6.8	44.2	4.5	40.1	2.5	36.5	0.0	32.0
12	53.6	10.5	50.9	8.5	47.3	6.8	44.2	4.3	39.7	2.0	35.6
14	57.2	12.3	54.1	10.5	50.9	8.5	47.3	6.2	43.2	3.7	38.7
16	60.8	14.4	57.9	12.6	54.7	10.5	50.9	8.3	46.9	5.6	42.1
18	64.4	16.5	61.7	14.6	58.3	12.4	54.3	10.0	50	7.4	45.3
20	68.0	18.3	64.9	16.5	61.7	14.4	57.9	11.9	53.4	9.2	48.6
22	71.6	20.3	68.5	18.4	65.1	16.3	61.3	13.7	56.7	11.6	52.8
24	75.2	22.2	72.1	20.5	68.9	18.4	65.1	15.6	60	13.0	55.4
26	78.8	24.4	75.9	22.2	72.1	20.1	68.2	17.6	63.6	14.7	58.5
28	82.4	26.3	79.3	24.2	75.6	22	71.6	19.5	67.1	17.5	63.5
30	86.0	28.3	82.9	26.3	79.3	23.9	75.0	21.5	70.7	18.3	64.9

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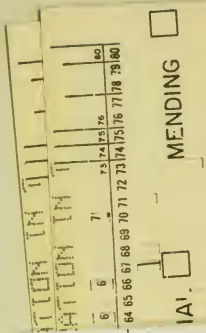

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